

**MINISTRY OF HIGHER EDUCATION AND SCIENTIFIC
RESEARCH**



Basrah University for Oil and Gas
College of Oil and Gas Engineering
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Power Generation Combined Cycle

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A Project Submitted in Partial Fulfillment of the Requirements for
the Degree of (B.Sc.) in Chemical and Petroleum Refining
Engineering

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Prepared by:

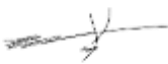
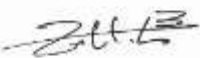
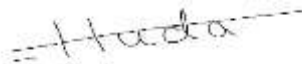
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ABSTRACT:

Combined Cycle Power Plant or combined cycle gas turbine, The gas turbine generator produces electricity, and the waste heat is used to generate steam through a steam turbine to generate more electricity. Gas turbine is one of the most efficient equipment to convert gas fuel into mechanical or electrical power.

It is also common to use distillate condensate fuels, usually diesel, as an alternative fuel.

Recently, as the efficiency of simple cycles has improved and the price of natural gas has decreased, gas turbines have been widely adopted for base load electricity generation, especially in the combined cycle, where waste heat that In steam boilers, recycling and steam are also used to generate additional electricity.

This system is known as a combined cycle.

The basic principle of the combined cycle is simple: burning gas in a gas turbine (Not GT Not only does it produce power - which can be converted into electrical power by an accompanying generator - but the hot exhaust gases from its chimney can also be used.

This type of power plant has been installed in large numbers around the world, in places where there is access to significant amounts of natural gas. The combined cycle power plant produces high power outputs at high efficiency (up to 55%) with low greenhouse gas emissions.

While in a power plant Normally We get 33% of electricity and the remaining 76% is wasted. Using a combined cycle power plant 76% of electricity is obtained.

الخلاصة:

محطة توليد الطاقة بدورة متكاملة أو محطة التوربينات الغازية بدورة متكاملة، يقوم مولد التوربين الغازي بإنتاج الكهرباء، ويتم استخدام الحرارة الناتجة كنتاج جانبي منه لتوليد البخار ومن ثم إلى توربين بخاري ليتم توليد المزيد من الكهرباء. التوربين الغازي هو أحد أكثر الأجهزة كفاءةً لتحويل الوقود الغازي إلى طاقة ميكانيكية أو كهربائية. وفي الآونة الأخيرة، مع تحسين كفاءة الدورات البسيطة وانخفاض سعر الغاز الطبيعي، تم اعتماد التوربينات الغازية على نطاق واسع لتوليد الكهرباء ذات الحمولة الأساسية، خاصة في الدورة المتكاملة، حيث يتم استخدام حرارة النفايات في مراحل البخار، وإعادة تدوير البخار لتوليد الكهرباء الإضافية، وتُعرف هذه النظام بدورة متكاملة أو مشتركة.

المبدأ الأساسي للدورة المتكاملة بسيط: حرق الغاز في توربين غازي (وهذا يعني أنها ليست فقط تنتج طاقة والتي يمكن تحويلها إلى طاقة كهربائية من خلال مولد) ولكن يمكن أيضاً استخدام الغازات الساخنة من مدخنته لتوليد الكهرباء.

تم تثبيت هذا النوع من محطات الطاقة بأعداد كبيرة حول العالم، في الأماكن التي تتوفر فيها كميات كبيرة من الغاز الطبيعي . .

تنتج هذه المحطة مخرجات طاقة عالية بكفاءة عالية (تصل إلى 55%) مع انبعاثات منخفضة من الغازات الدفيئة. بينما في محطة توليد الطاقة عادةً ما نحصل على 33% من الكهرباء ويتم إضاعة النسبة المتبقية 76%. باستخدام محطة توليد الطاقة بدورة متكاملة يتم الحصول على 76% من الكهرباء..

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Chapter One

Introduction

INTRODUCTION

1.1 History of Energy Flows

Human beings, like other living organisms, require energy to live. Over the last two hundred years, people in industrialized countries have experienced an unprecedented shift in the nature and scale of their energy consumption. the shift from the organic energy regime of muscle power, wood, wind, and water power to the mineral regime of coal, oil, gas, and electricity has allowed people to do a wide range of things previously difficult or even impossible. The cheap and abundant energy created by the energy revolution has made it possible to have more heat, light, and transportation, enormously reducing the amount of hard physical labor that so many people endured throughout their lives. By creating previously unknown levels of economic growth, the new energy regime has also increased the standard of living in many nations to new heights. [1]

Electricity will let you live like a King!” proclaimed a typical advertisement from the 1920s, “and it was not far wrong: only a tiny minority of people throughout history have previously had access to the kind of wealth now provided by modern forms of energy. These influence every aspect of daily life, from the food people eat to the places they live and work, from the way they get from place to place to the way they relate to others. [2]

1.1.1 The organic energy regime:

The discovery and control of fire by *Homo erectus*, dating back at least 500,000 years represented the first great ecological transition for humans, influencing their evolutionary trajectory. "The discovery of fire and the exploitation of draft animals marked two main changes in the history of technology. Although scant

evidence exists about how this discovery and use were made, watching naturally occurring fires caused by volcanic eruptions and falling rocks may provide some of the circumstantial evidence. [3]

The invention of the plow, the wheel, the cart, and much else accelerated the transition from human to animal energy for work. Population growth became one of its unintended consequences. [3]

For archaic humans, the primary energy source derived from their labor of gathering and scavenging and then transitioning to overhunting large megafauna and eventually smaller mammals. They consumed nuts, fruits, grains, edible vegetables, animal flesh, and fish, converting the food into energy for work. Food is to humans and animals and gasoline is to mechanical engines. The major difference between humans and machines, however, requires humans to expend energy in search of the food that they need to survive. Archaic humans spent the majority of their waking hours securing the food that they needed to survive. If they failed in this quest because of injury, poor judgment, an inferior skill set, or environmental conditions, they died. [3]

Over time, successful technical innovations, including throwing projectiles, such as clubs, sharpened spears, bows, and arrows, and strategies for corralling prey for the kill increased efficiency. At the same time, overhunting and burning off the vegetation to make the hunt more successful could alter the environment in fundamental ways. Regardless, their food base remained very limited and populations were therefore extremely sparse. [3]

1.1.2 The mineral energy regime

1.1.2.1 Coal

Coal's transportability and its availability signaled a shift from an organic economy of muscle, wood, wind, and water to a mineral economy that would initially be dominated by coal production and consumption but eventually include petroleum and natural gas use. Coal is the product of solar energy formed by decaying giant tree ferns and other vegetation that flourished in the great hot delta swamps of the Paleozoic Era, some 300 million years ago. [2]

1.1.2.2 Petroleum

Microscopic organisms contain more hydrogen molecules than coal and can more easily be transformed under pressure into hydrocarbons. Crude oil is a combination of liquid and gaseous hydrocarbons. Once refined, the former becomes kerosene, gasoline, and semisolid asphalt. Refined gaseous hydrocarbons become 4 Petroleum “Liquid gold” Petroleum 85 propane, butane, methane or natural gas. Gaseous hydrocarbons exist as billions of bubbles making the liquid crude oil percolate and press against the miles of sediment that became solid sandstone. Some of it leaked through cracks spilled onto the ground and released natural gas into the atmosphere. [2]

1.1.2.3 gas and natural gas

Manufactured gas became one of the first networked production and distribution enterprises along with water and sewer systems of nineteenth-century cities. It was preceded by decades of the development of technological systems such as trolleys, telephones and electric systems to create an integrated infrastructure. [2]

1.1.2.4 Nuclear Power

On December 2, 1942, man achieved here the first self-sustaining chain reaction and thereby initiated the controlled of nuclear energy.” In the aftermath of World War

In 1945, Lewis L. Strauss, chairman of the newly created Atomic Energy Commission (AEC), expressed optimism about the future of nuclear energy succinctly. He claimed, without qualification, “It is not too much to expect that our children will enjoy electrical energy in their homes too cheap to meter.” The civilian use of nuclear power captured the attention of government officials and private utilities. Scientists hypothesized that 1 ton of uranium by fission would release as much energy as 3 million tons of coal.

1.1.3 The renewable energy regime

The renewable energy regime refers to the set of policies, regulations, and laws that promote the use of renewable energy sources such as solar, wind, hydro, and

geothermal power. It aims to reduce dependence on non-renewable energy sources and mitigate the negative impacts of climate change. The regime also encourages the development of innovative technologies and practices to increase the efficiency and affordability of renewable energy systems. [2]

1.1.3.1 Hydropower

During the twentieth century, governments and private utilities have transformed their relationship with rivers and their tributaries by building as many as 45,000 large hydroelectric dams. Proponents argued that with electricity usage projected to nearly double by 2035 from 5.2 terawatt-hours to 9.3 terawatt-hours, big projects such as large hydroelectric dams offered many benefits.¹ In addition to meeting electrical needs, the benefits, according to proponents, included a reduction in fossil fuel consumption, flood control, irrigation, improved water transportation, and employment for the construction industry. [2]

1.1.3.2 Solar Power

Solar power became the world's leader in new electricity generation in 2017. Worldwide, public utilities and private companies installed 73 gigawatts (1 gigawatt is 1 billion watts) of new solar photovoltaic capacity. The wind followed with 55 gigawatt-hours of new capacity, coal with 52 gigawatt-hours, natural gas with 37 gigawatt-hours, and hydroelectric with 28 gigawatt-hours. However, its contribution to the world's energy supply recently passed 1 percentage point. Installed capacity remains dominated by fossil fuels with oil at 33 percent, coal at 30 percent, natural gas at 24 percent, hydro at 7 percent, nuclear at 4 percent, and solar and wind at 2 percent. Technological advances in production resulted in significant declines in the price of solar panels with China becoming the world's primary supplier. Additionally, the Sun, recognized as Earth's source of light and heat now became scientifically acknowledged as the galaxy's vast nuclear fusion reactor. In effect, Earth receives 885 million terawatt-hours each year, or 165,000 terawatts (1 terawatt equals 1 million megawatts) of power from the Sun every moment of every day. [2]

1.1.3.3 Power of Wind

Sources of energy change with fluctuating needs, with availability, costs and the density to produce power efficiently. The power of the wind, however, has remained a constant throughout the history of the planet. It made placid seas churn with violent waves. Its force swept across the land, rearranging the topography, creating dunes in some places and eroding hilltops in others. Creation and erosion

signaled its power and presence. Dust storms became a signature event by moving topsoil across continents. In human history, its energy powered filled sails and the blades of early windmills to raise groundwater to the surface and grind grain into flour. Over time the increasing size and efficiency of windmills assumed many more functions. The invention of turbines, a mechanical breakthrough, set the stage for the modern technology of wind power. [2]

1.1.4 Alternative energy solutions

1.1.4.1 Fuel cells and battery power:

Two possible energy sources to curtail greenhouse gas emissions are hydrogen fuel cells and storage batteries for vehicles and the electrical grid. Using gasoline to power vehicles emits 20 pounds of carbon dioxide for each gallon of gasoline burned.¹ It is the most energy inefficient use of fossil fuel. For every twenty gallons put into a car we only get two gallons of actual work. The rest is dissipated as heat. The other inherent inefficiency is in the fact to move one person, we have to spend energy moving over a ton of material (the automobile itself). Fuel cells convert the chemical energy found in hydrogen into electrical energy in one step.

Improvements in storing the energy from sunlight and wind for the stationary electrical grid using battery power will reduce emissions as well. Electrical Energy Storage (EES) converts electrical energy into a stored form that can later be converted when needed into electricity. Globally in 2018, installed energy storage reached 175.8 gigawatt-hours. In Europe, 10 percent of its energy was provided by energy storage facilities, and in Japan, it reached 15 percent. EES technologies described here focus on electrical grids. They include storage technology types, including Pumped Hydroelectric Storage (PHES) and Compressed Air Energy Storage (CAES). Several other battery storage projects include those based on lead-acid, lithium-ion, nickel and sodium-based and flow batteries. [2]

1.1.4.2 Hydrogen fuel cells

As a replacement for gasoline vehicles, scientists and engineers project that fuel cell vehicles will become two to three times more efficient. Relative to other ways to generate electricity, fuel cells have the advantages of low costs with high efficiencies. Simply stated, a fuel cell takes oxygen from the air and hydrogen from a tank, creating a chemical reaction to produce water vapor and electric

energy. Typically, fuel cells burn hydrogen that reacts with oxygen in the air to produce electricity and water. Neither produces heat-trapping gases that warm the global climate system. Since hydrogen-powered fuel cells do not pollute, they offer one potential solution to curbing environmental pollution. Hydrogen-powered fuel cell vehicles with a battery and an electric motor could double automotive fuel efficiency. Producing hydrogen through low-carbon-emitting processes and establishing a functional distribution system, comparable to the network of existing gas stations, would make possible low-greenhouse emissions hydrogen fuel cell vehicles. Achieving such outcomes will require technological innovations and years before fuel cell vehicles become commonplace. [2]

1.2 Overview Of Electrical Power Generation

Electrical energy is produced by converting sources of energy available in nature. Various energy sources available in nature are the Sun, wind, water, fossil fuels, and nuclear fuels. Electrical power generation can be broadly classified as conventional or non-renewable power generation, and non-conventional or renewable power generation [2]

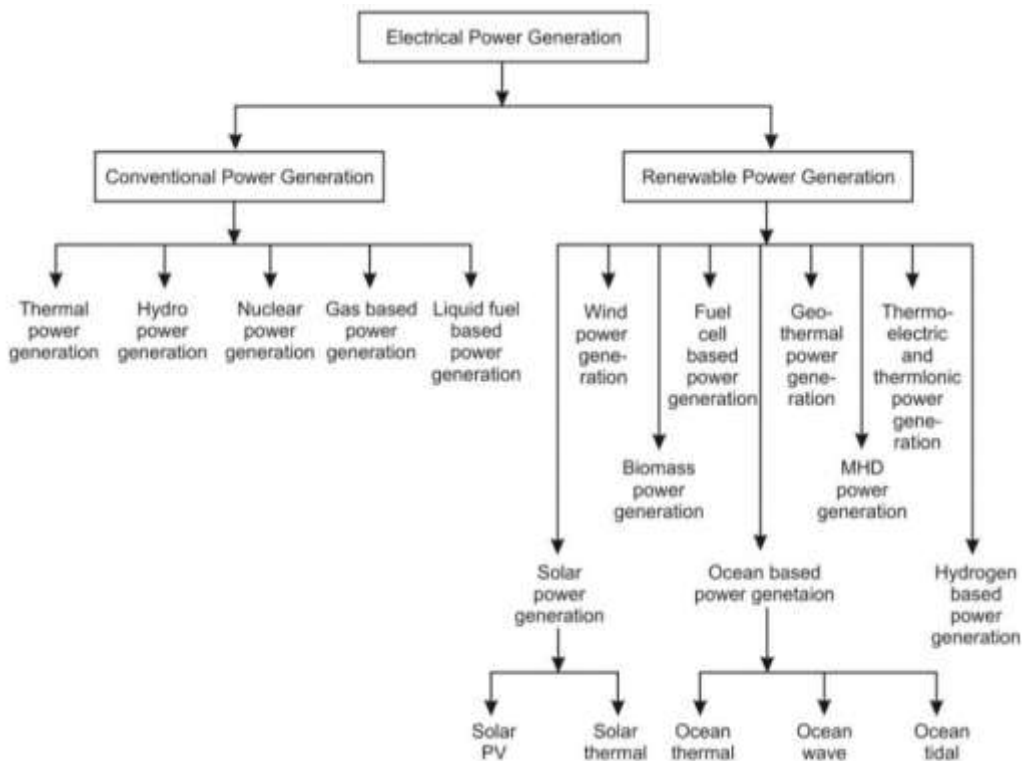


Figure 1.1

1.3 Types Of Turbines In Power Plants:

1.3.1 Wind Turbine: A wind turbine, is a machine that converts the power in the wind into electricity.

As electricity generators, wind turbines are connected to some electrical network. These networks include battery-charging circuits, residential-scale power systems, isolated or island networks, and large utility grids. In terms of total numbers, the most frequently found wind turbines are quite small. on the order of 10 kW or less. This contrasts with a ‘windmill’, which is a machine that converts the wind’s power into mechanical power. History of Wind Energy The first known historical reference to a windmill is from Hero of Alexandria, in his work Pneumatics (Woodcroft, 1851). Hero was believed to have lived either in the 1st century B.C. or the 1st century A.D. [4]

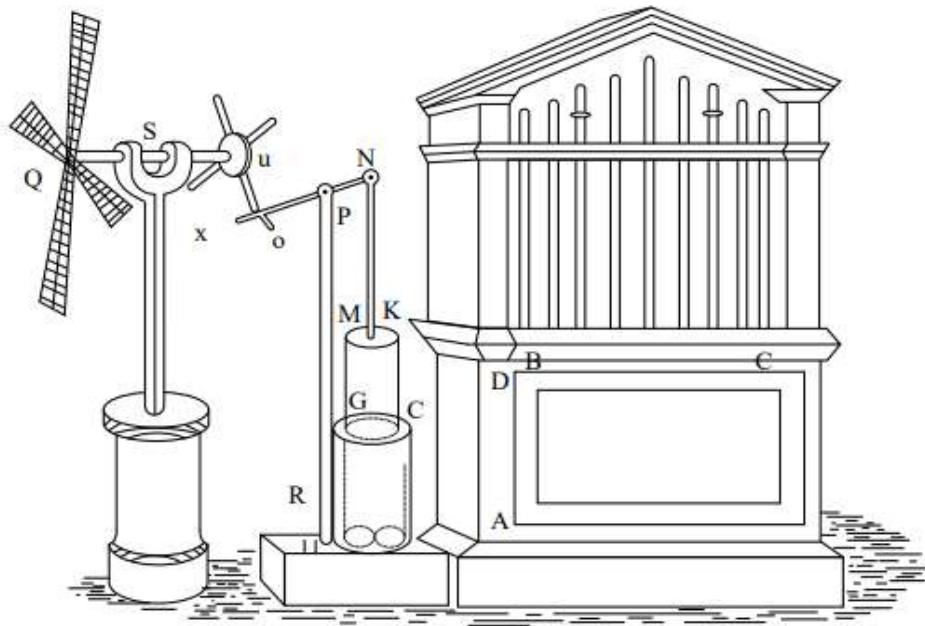


Figure 2 Hero's windmill (from woodcroft ,1851

)

Components of a Wind Turbine Wind turbine usually has six main components: the rotor, the gearbox, the generator, the control and protection system, the tower, and the foundation. [5]

1.3.2 Gas Turbine

A gas turbine engine is composed of three major sections are Compressors, Combustion chambers, and Turbine wheels. Air is taken in through the air inlet duct by the compressor which raises pressure and temperature. The air is then discharged into the combustion chamber(s) where fuel is admitted the fuel nozzle(s).

The fuel-air mixture is ignited by ignitor(s) and combustion takes place. Combustion is continuous, and the ignitors are de-energized after a brief period. The hot and rapidly expanding gases are directed towards the turbine rotor assembly. Kinetic and thermal energy are extracted by the turbine wheels. The action of the gases against the turbine blades causes the turbine assembly to rotate. The turbine rotor is connected to the compressor which rotates with the turbine. The exhaust gases then are discharged through the exhaust duct. Since approximately 75% of the power developed by a gas turbine engine is used to drive the compressor and accessories, only 25% can be used to drive a generator or to propel a ship. [6]

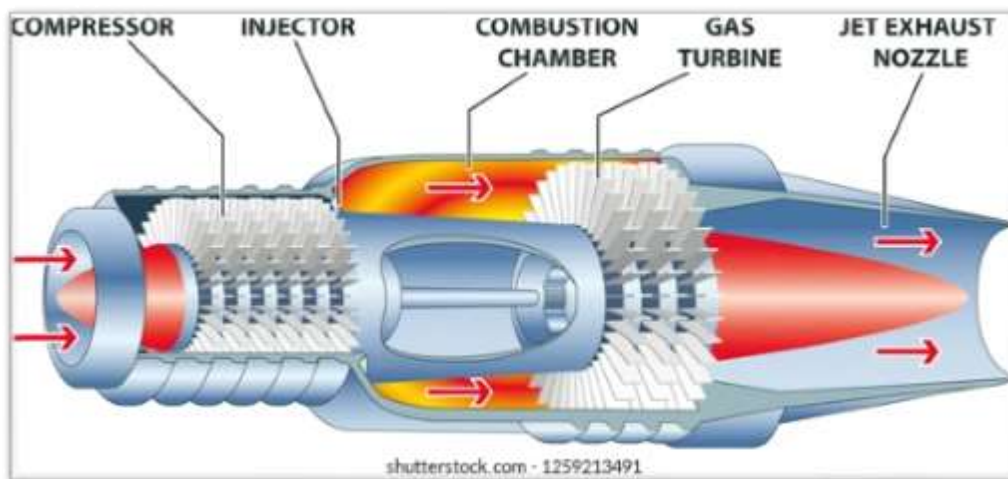


Figure 3 gas turbine

1.3.3 Water Turbine

the turbine or water wheel is a very ancient device. It was in use in Asia Minor for about 2,500 years, but it was not until about 80 years ago that any serious attempts were made to place the scientific theory upon a proper basis and to construct turbines in accordance with it. During the last 20 years, great modifications and improvements have been made, and there is now no other prime mover of which the theoretical laws are more definite and clearly understood. [7]

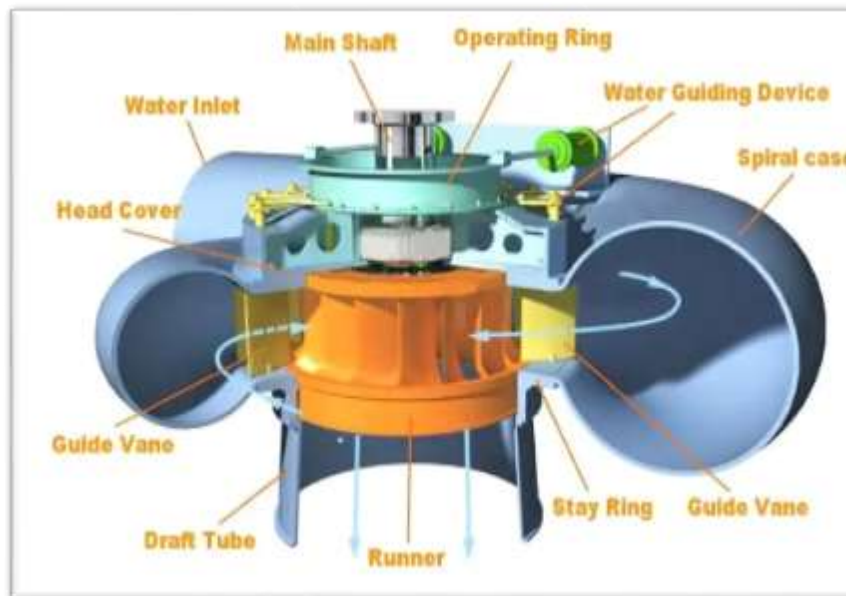


Figure 4 water turbine

1.3.4 Steam Turbine

The steam turbine is a prime-mover for generating electricity. In which the potential energy of the steam is transformed into kinetic energy, and latter in its turn is transformed into the mechanical energy of rotation of the turbine shaft. The turbine shaft, directly or with the help of a reduction gearing, is connected with the driven mechanism. Depending on the type of the driven mechanism a steam turbine may be utilised in most diverse fields of industry, for power generation and for transport.

Transformation of the potential energy of steam into the mechanical energy of rotation of the shaft is brought about by different means.

In the turbine, the energy level of the working fluid goes on decreasing along the flow stream. Single unit of steam turbine can develop power ranging from 1mW to 1000mW.

The purpose of turbine technology is to extract the maximum quantity of energy from the working fluid, to convert it into useful work with maximum efficiency, by means of a plant having maximum reliability, minimum cost, minimum supervision and minimum starting time. [8]

1.3.4.1. Principle Of Operation Of Steam Turbine

The principle of operation of steam turbine is entirely different from the steam engine. In reciprocating steam engine, the pressure energy of steam is used to overcome external resistance and the dynamic action of steam is negligibly small. But the steam turbine depends completely upon the dynamic action of the steam. According to Newton's Second Law of Motion, the force is proportional to the rate of change of momentum ($\text{mass} \times \text{velocity}$). If the rate of change of momentum is caused in the steam by allowing a high velocity jet of steam to pass over curved blade, the steam will impart a force to the blade. If the blade is free, it will move off (rotate) in the direction of force. In other words, the motive power in a steam turbine is obtained by the rate of change in moment of momentum of a high velocity jet of steam impinging on a curved blade which is free to rotate. The steam from the boiler is expanded in a passage or nozzle where due to fall in pressure of steam, thermal energy of steam is converted into kinetic energy of steam, resulting in the emission of a high velocity jet of steam which, Principle of working impinges on the moving vanes or blades of turbine (Fig. 3.1). [8]

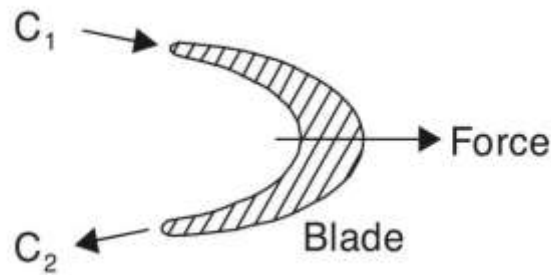


Figure 5 turbine blades

Attached on a rotor which is mounted on a shaft supported on bearings, and here steam under- goes a change in direction of motion due to curvature of blades which gives rise to a change in momentum and therefore a force.

This constitutes the driving force of the turbine. This arrangement is shown. It should be realized that the blade obtains no motive force from the static pressure of the steam or from any impact of the jet, because the blade is designed such that the steam jet will glide on and off the blade without any tendency to strike it.

As shown in Fig. 3.2, when the blade is locked the jet enters and leaves with equal velocity, and thus develops maximum force if we neglect friction in the blades. Since the blade velocity is zero, no mechanical work is done.

As the blade is allowed to speed up, the leaving velocity of jet from the blade reduces, which reduces the force. Due to blade velocity the work will be done and maximum work is done when the blade speed is just half of the steam speed. In this case, the steam velocity from the blade is near about zero i.e.

It is trail of inert steam since all the kinetic energy of steam is converted into work. The force and work done become zero when the blade speed is equal to the steam speed. From the above discussion, it follows that a steam turbine should have a row of nozzles, a row of moving blades fixed to the rotor, and the casing (cylinder). A row of nozzles and a row of moving blades constitutes a stage of turbine. [8]

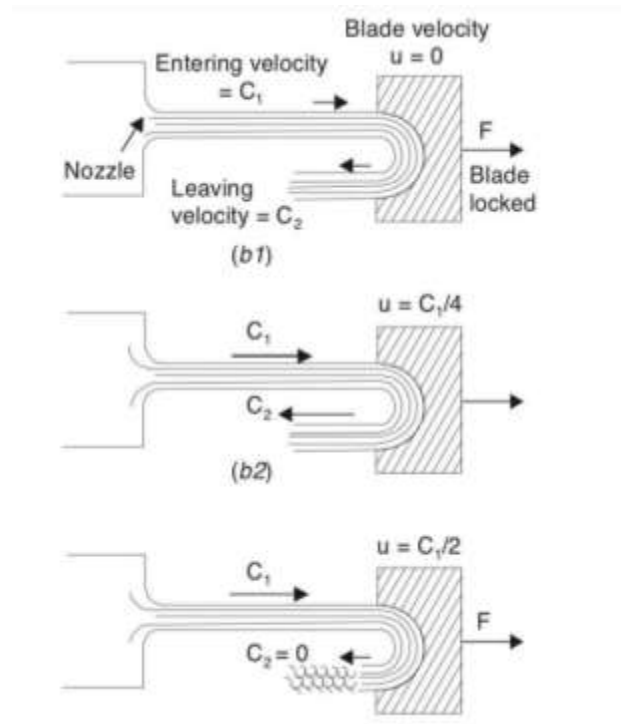


Figure 6 Action of jet on Blade

1.3.4.2. Classes of Steam Turbines

There are a several ways in which steam turbine may be classified, as

(A) On the basis of principle of operation

- 1- Impulse
- 2- Pure Reaction
- 3- Impulse-reaction
- 4- Combination

(B) On the basis of "Direction of Flow" :

1. Axial flow turbine
2. Radial flow turbine
3. Tangential flow turbine.

(C) On the Basis of Means of Heat Supply:

- 1- Single pressure turbine
- 2- Mixed or dual pressure turbine
- 3- Reheated turbine

(i) Single (ii) Double

(D) On the Basis of Means of Heat Rejection :

- 1- Pass-out or extraction turbine
- 2- Regenerative turbine
- 3- Condensing turbine,
- 4- Non- condensing turbine
- 5- Back pressure or topping turbine

(F) On the Basis of Number of Cylinder: Turbine may be classified as

1. Single cylinder
2. Multi-cylinder

(G) On the Basis of Arrangement of Cylinder Based on General Flow of Steam

1. Single flow
2. Double flow 3. Reversed flow 4. [8]

1.3.5 Impulse Turbine:

This type of turbine works on the principle of impulse and is shown diagrammatically. It mainly consists of a nozzle or a set of nozzles, a rotor mounted on a shaft, one set of moving blades attached to the rotor and a casing. The uppermost portion of the diagram shows a longitudinal section through the upper half of the turbine, the middle portion shows the development of the nozzles and blading i.e. the actual shape of the nozzle and blading, and the bottom portion shows the variation of absolute velocity and absolute pressure during flow of steam through passage of nozzles and blades. The example of this type of turbine is the de-Laval Turbine.

It is obvious from the figure that the complete expansion of steam from the steam chest pressure to the exhaust pressure or condenser pressure takes place only in one set of nozzles i.e. the pressure drop takes place only in nozzles. It is assumed that the pressure in the recess between nozzles and blades remains the same. The steam at condenser pressure or exhaust pressure enters the blade and comes out at the same pressure i.e. the pressure of steam in the blade passages remains approximately constant and equal to the condenser pressure. Generally, converging-diverging nozzles are used. Due to the relatively large ratio of expansion of steam in the nozzles, the steam leaves the nozzles at a very high

velocity (supersonic), of about 1100 m/s. It is assumed that the velocity remains constant in the recess between the nozzles and the blades. The steam at such a high velocity enters the blades and reduces along the passage of blades and comes out with an appreciable amount of velocity (Fig3.3). [9]

As it has been already shown, that for the good economy or maximum work, the blade speed should be one half of the steam speed so blade velocity is of about 500 m/s which is very high. This results in a very high rotational speed, reaching 30,000 r.p.m. Such high rotational speeds can only be utilised to drive generators or machines with large reduction gearing arrangements [9]

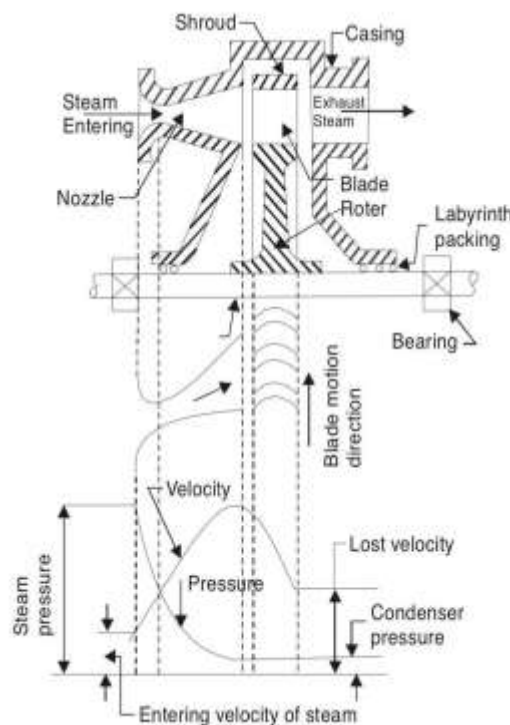


Figure 7 Impulse turbine

In this turbine, the leaving velocity of steam is also quite appreciable resulting in an energy loss, called “carry over loss” or “leaving velocity loss”. This leaving loss is so high that it may amount to about 11 percent of the initial kinetic energy. This type of turbine is generally employed where relatively small power is needed and where the rotor diameter is kept fairly small. [9]

Classification of Impulse Turbines

Returning to the first system of classification, turbines of Class I. may be subdivided into four groups as follows :—

- 1- simple or single stage impulse
- 2- Velocity Compounded Impulse.
- 3- Pressure-Velocity Compounded Impulse.
- 4- Pressure Compounded Impulse.

1.3.5.1 Pressure Compounded Impulse Turbine.

In this type of turbine, the compounding is done for pressure of steam only. to reduce the high rotational speed of turbine the whole expansion of steam is arranged in a number of steps by employing a number of simple turbine in a series keyed on the same shaft as shown. Each of these simple impulse turbine consisting of one set of nozzles and one row of moving blades is known as a stage of the turbine and thus this turbine consists of several stages. The exhaust from each row of moving blades enters the succeeding set of nozzles. Thus we can say that this arrangement is nothing but splitting up the whole pressure drop (Fig. 3.4).

[9]

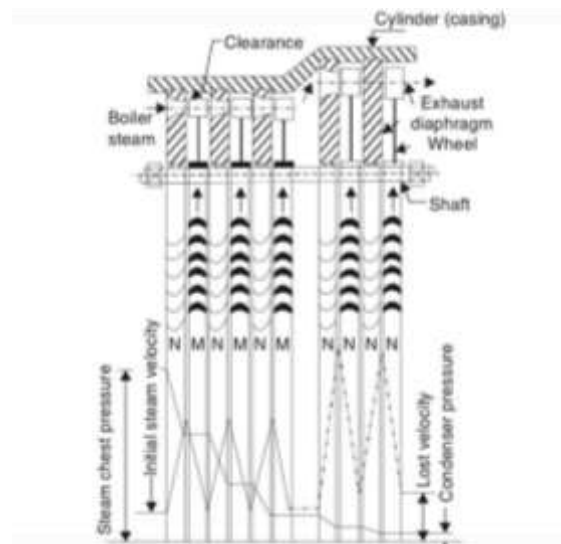


Figure 8 PRESSURE COMPOUNDED IMPULSE TURBINE

1.3.5.2 Impulse-Reaction Turbine:

As the name implies this type of turbine utilizes the principle of impulse and reaction both. Such a type of turbine is diagrammatically shown. There are a number of rows of moving blades attached to the rotor and an equal number of fixed blades attached to the casing. [9]

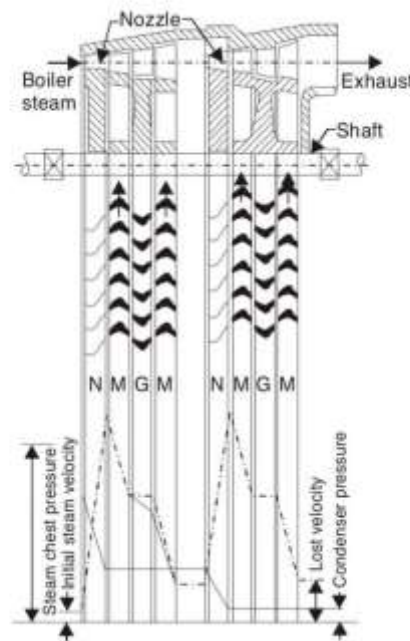


Figure 9 Impulse-Reaction Turbine

1.3.6 Axial Flow Turbine:

In axial flow turbine, the steam flows along the axis of the shaft. It is the most suitable turbine for large turbo-generators and that is why it is used in all modern steam power plants. [10]

1.3.7 Radial Flow Turbine:

In this turbine, the steam flows in the radial direction. It incorporates two shafts end to end, each driving a separate generator. A disc is fixed to each shaft. Rings of 50% reaction radial-flow bladings are fixed to each disk. The two sets of bladings rotate counter to each other. In this way, a relative speed of twice the

running speed is achieved and every blade row is made to work. The final stages may be of axial flow design in order to achieve a larger area of flow. Since this type of turbine can be warmed and started quickly, so it is very suitable for use at times of peak load.

Though this type of turbine is very successful in the smaller sizes but formidable design difficulties have hindered the development of large turbines of this type. In Sweden, however, composite radial/axial flow turbines have been built of outputs upto 275 MW. Sometimes, this type of turbine is also known as Liungstrom turbine after the name of its inventor B and F. Liungstrom of Sweden (Fig. 3.5). [10]

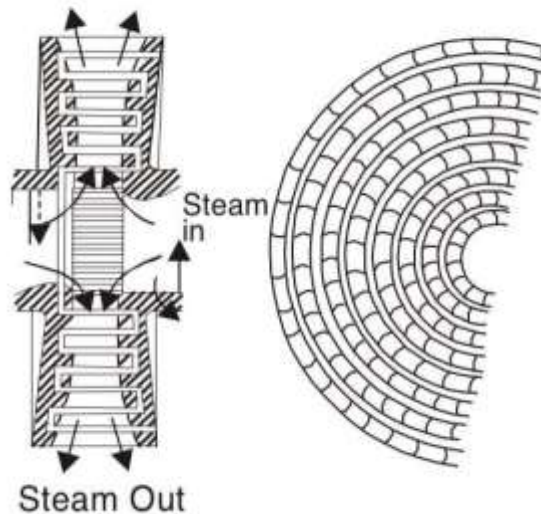


Figure 10 Tangential Flow

In this type, the steam flows in the tangential direction. This turbine is very robust but not particularly efficient machine, sometimes used for driving power station auxiliaries. In this turbine, nozzle directs steam tangentially into buckets milled in the periphery of a single wheel, and on exit the steam turns through a reversing chamber, reentering bucket further round the periphery. This process is repeated several times, the steam flowing a helical path. Several nozzles with reversing chambers may be used around the wheel periphery. [10]

1.3.8 Single Pressure Turbine:

In this type of turbine, there is single source of steam supply.

1.3.9 Mixed or Dual Pressure Turbine:

This type of turbines, use two sources of steam, at different pressures. The dual pressure turbine is found in nuclear power stations where it uses both sources continuously. The mixed pressure turbine is found in industrial plants (e.g., rolling mill, colliery, etc.) where there are two supplies of steam and use of one supply is more economical than the other; for example, the economical steam may be the exhaust steam from engine which can be utilised in the L. P. stages of steam turbine. Dual pressure system is also used in combined cycle. [10]

1.3.10 Reheated Turbine:

During its passage through the turbine steam may be taken out to be reheated in a reheater incorporated in the boiler and returned at higher temperature to be expanded in (Fig. 3.3). This is done to avoid erosion and corrosion problems in the bladings and to improve the power output and efficiency. The reheating may be single or double or triple. [10]

1.3.4.3 Top Steam Turbine Manufacturer in the World in 2022

Company	Headquarters	No. of Employees	Annual Sales
<u>Bradken, Inc. (Engineered Products Business)</u>	Kansas City, MO	1000+	\$250 Mil. and over
<u>Mitsubishi Heavy Industries America, Inc.</u>	New York, NY	1000+	\$250 Mil. and over
Turboatom	Kharkov, Ukraine	1000+	\$250 Mil. and over
<u>Bharat Heavy Electricals Limited (BHEL)</u>	Delhi, India	1000+	\$250 Mil. and over
TURBOPAR Group	Smolensk, Russia	100-200	NA

5/10

Company	Headquarters	No. of Employees	Annual Sales
Mitsubishi Hitachi Power Systems Europe, Ltd.	Duisburg, Germany	1000+	\$250 Mil. and over
<u>Siemens Energy</u>	Munich, Germany	1000+	\$250 Mil. and over
<u>General Electric Co.</u>	Boston, MA	1000+	\$250 Mil. and over
<u>Dongfang Electric Corp Ltd</u>	Sichuan, China	1000+	\$250 Mil. and over
Sande Stahlguss GmbH	Sande, Germany	200-499	NA

Bradken, Inc. (Engineered Products Business)

Bradken is a metal casting and foundry firm based in the United States that specializes in custom-designed iron and steel parts and equipment. The world's largest mining businesses rely on this company for customized products such as ground engaging tools (GET), crawler shoes, mill liners, crusher liners, wear solutions, and monitoring systems. In addition, they provide specialist structural and industrial casting services in a variety of alloys, with casting capacities ranging from 500 grams to more than 25 tons. Bradken has sales and service teams all around the world, as well as foundries and workshops. They employ approximately 3,000 people in 20 production facilities and 40 sales and service centers across Australia, the United States, India, Indonesia, New Zealand, Canada, Malaysia, South Africa, China, and South America. They also manufacture custom turbines, including steam turbines. Steam turbines are used in a variety of applications, including process equipment, power generation, hydro, military, mining, construction and industrial machinery, shipbuilding, and transportation.[10]

Mitsubishi Heavy Industries America, Inc.

supports aircraft and aerospace projects, including the MU-2 and MU-300, at their long-time approved Mitsubishi Service Center, Intercontinental Jet Service Corp. in Tulsa, Oklahoma. Mitsubishi Heavy Industries America Corrugating Machinery Division also provides knowledgeable equipment systems, sales, parts, technical, and field support to the developing, dynamic packaging industry, making it one of the most respected converter providers in North and Central America. Their patented post-combustion capture technologies allow industrial and power plants to absorb and retain up to 90% of CO₂ emissions, putting them at the forefront of green technology adoption. It is how they make a positive difference in the world [10]

Since 1991, the Tire Machinery Division of Mitsubishi Heavy Industries America, Inc. (MHIA) has been manufacturing and assembling tire industry equipment in Northeast Ohio. MHIA works with Mitsubishi Heavy Industries, Ltd. (Japan) and Mitsubishi Heavy Industries Machinery Technology Corporation to provide worldclass designs, engineering, and equipment to the tire industry. Mitsubishi Heavy Industries America, Inc.'s Transportation Systems Division is

in charge of the design, development, supply, engineering, marketing, and installation of

innovative transit systems for the global market. In addition to their world-class Automated People Mover (APM) systems, they have a diverse product portfolio that includes rail transit, maglev, monorail, and short-haul transportation systems that can all be customized for practically any application. [10]

1.4 Combined Cycle Power Plant

1.4.1 Introduction

A combined cycle power plant is a power plant that includes a number of gas turbines and steam turbines. In this type of power plant, using a recovery boiler, the heat in the exhaust gases from the gas turbines is used to produce the water vapor needed in the steam turbines. If the turbine gas to If there is no combined cycle, its output gases, which can have a temperature of up to 660 degrees Celsius, enter the air directly and the remaining energy is wasted. While in the combined cycle power plant, this energy is used and the steam turbine boiler is not needed. It produces steam as fuel. Therefore, by using this method, the efficiency of the cycle increases. Combined cycle reservoir sit is a very efficient, flexible, reliable, cost-effective and environmentally friendly solution for power generation. Power plant (Combined cycle) The combined cycle is actually a combination of a steam turbine and a gas turbine in such a way that the gas turbine generator produces electricity, at the same time, the thermal energy wasted from the gas turbine (by combustion products) is used to produce the steam required by the steam turbine. in this way additional electricity is produced. By combining these two cycles, the efficiency of the power plant increases. It has 25 to 40 percent, while the same power plant with Combined cycle electrical efficiency approx. It has 60%. As mentioned, these power plants are made from a combination of steam and gas turbines and depending on the type of turbines, heat recovery boilers, and recovery devices, there are many types. By using gas turbines in combined cycles, its low efficiency can be eliminated and as a result, it can be used to supply base load was used, while it also benefited from its other advantages such as fast start-up and its flexibility in a wide range of loads. Theoretically, the energy that can be recovered from the exhaust of gas turbines is about half of the energy produced by The turbine itself is gas. Therefore, the power of the steam turbine will be about half of the gas turbine. In some designs, two gas turbines create the energy required for one steam turbine, and as a result, the production power of steam

turbines is about that of gas turbines. The idea of the combined cycle was proposed to improve the efficiency of the simple Brayton cycle by using the heat of the gas turbine exhaust gases. This was tested by heat recovery. The heat recovery could recover the energy that was wasted from the gas turbine output. 70 to 60 the percentage of given energy. The heat exchanger cannot increase the output power and only increases the efficiency. Since the heat exchanger introduces a large pressure drop into the cycle, its use reduces the pressure ratio of the turbine and The result is a decrease in net output power. Due to the maximum power of simple cycles, they are used in places where the output efficiency is less important. while recovery cycles are used in cases where high efficiency is needed. As a result, the output power of the recovery cycle is about 11 to 14% lower than the simple cycle, which in a general evaluation we conclude that the efficiency of the gas turbine power plant It is a costly method along with the regenerator. Therefore, one should look for a method through which both requirements can be achieved, i.e. high efficiency and power.[11]

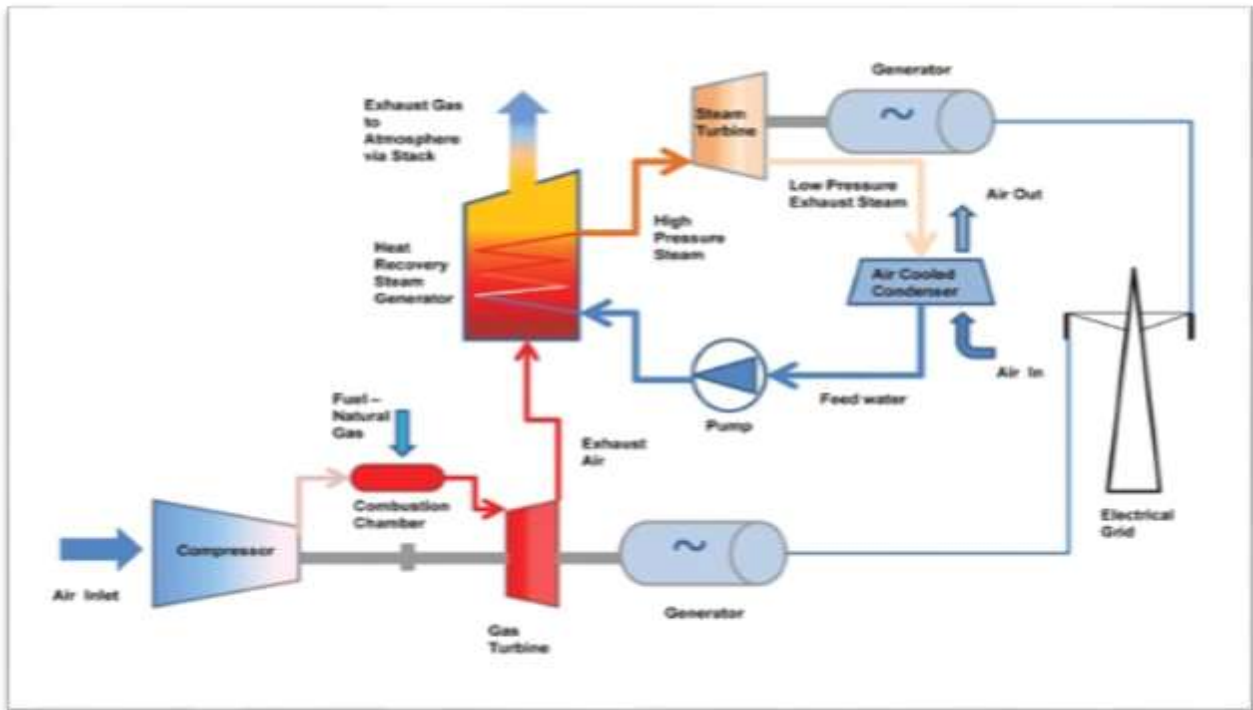


Figure 11 Schematic of combined cycle

The proposed solution is actually using the very high thermal energy of the gas. The output of the gas turbine was to produce the steam needed by the steam power plant. The gas turbine has gases with a temperature of approx. 1200 to 1600°C, and gas turbine engine with temp About 530 to 640 degrees Celsius, which by simultaneously combining the gas turbine on the hot side and the steam turbine on the cold side makes the power plant. It is called a combined cycle. The first combined cycle power plant was built in 1950. Since then, the number of combined cycle power plants, especially in the decade in 1970, it increased rapidly. [11]

1.4.2 Types of combined cycle power plants

Combined cycle power plants are divided into the following categories according to the type of turbines and regenerators and the presence of burners:

- Combined cycle power plants with burners.
- Combined cycle power plants without burners.
- combined cycle power plants with heat recovery boilers equipped for recovery or heating of feed water.
- combined cycle power plants with multiple steam pressure heat recovery boiler.
- combined cycle power plants with closed cycle gas turbine with heating of feed water in steam cycle.

In the first type of power plants, burners are placed inside the boiler and it is mostly used in power plants where the steam part is supposed to work permanently, in which case it should not depend on the gas turbine. In the second type of these power plants, hot gases are used, which is used as combustion products from the gas turbine. This exhaust smoke has a high volume and a temperature of about 500 °C and into the boiler is sent to convert water into steam so that the steam energy is used to drive the generator. The use of different types of combined cycle is different. The combined cycle plant without burner is mostly used to supply the base and intermediate load. In the third type of these power plants in the combined cycle, the exhaust gases of a simple gas turbine cycle that includes an air compressor) Room, (AC ignition) and is used there to produce

superheated steam (HRB), enters the heat recovery boiler (GT) and gas turbine (CC) In combined cycles that have low power, steam turbine power is approximate 50% less than the gas turbine Fourth, in these power plants, where steam is produced with multiple pressures, the temperature of the exhaust gases of the heat recovery boiler decreases, and in this way, the efficiency of the power plant increases in general. The simplest type of this cycle is the cycle with double pressure, although that the cycle with triple pressure is also used. For example, in a cycle with double pressure, the heat recovery boiler has two circuits to produce steam. The first circuit is the high-pressure circuit in which the steam produced enters it from the turbine inlet duct, and the second circuit is the low-pressure circuit in which the steam produced enters the turbine from the lower pressure stages. In a proposed combined cycle with Triple pressure, another steam is produced with a pressure between the input pressures to two steam turbines. This steam is injected into the combustion chamber of the gas turbine to reduce the emission of nitrogen oxides to the set standard. If this method used, some water will be lost which must be constantly compensated. .[11]

1.4.3 Mechanism of action

A combined cycle power plant, as the name suggests, provides a significant improvement in the thermal efficiency of a conventional steam power plant by combining existing gas and steam technologies. In a power plant CCGT, by piping the exhaust gas from the gas turbine to a heat recovery steam generator, the thermal efficiency is about It reaches 55-75 percent.

However, the heat recovered in this process is only enough to drive a steam turbine with an electrical output of approx. 55% of the turbine generator Enough gas. Gas turbine and steam turbine are coupled to a single generator. To set up, or operate "open loop" From the gas turbine alone, The steam turbine can be switched off using a hydraulic clutch. In terms of investing a single axis system approx. 5 percent less It costs, and this is due to the simplicity of its operation, which usually leads to higher reliability. .[11]

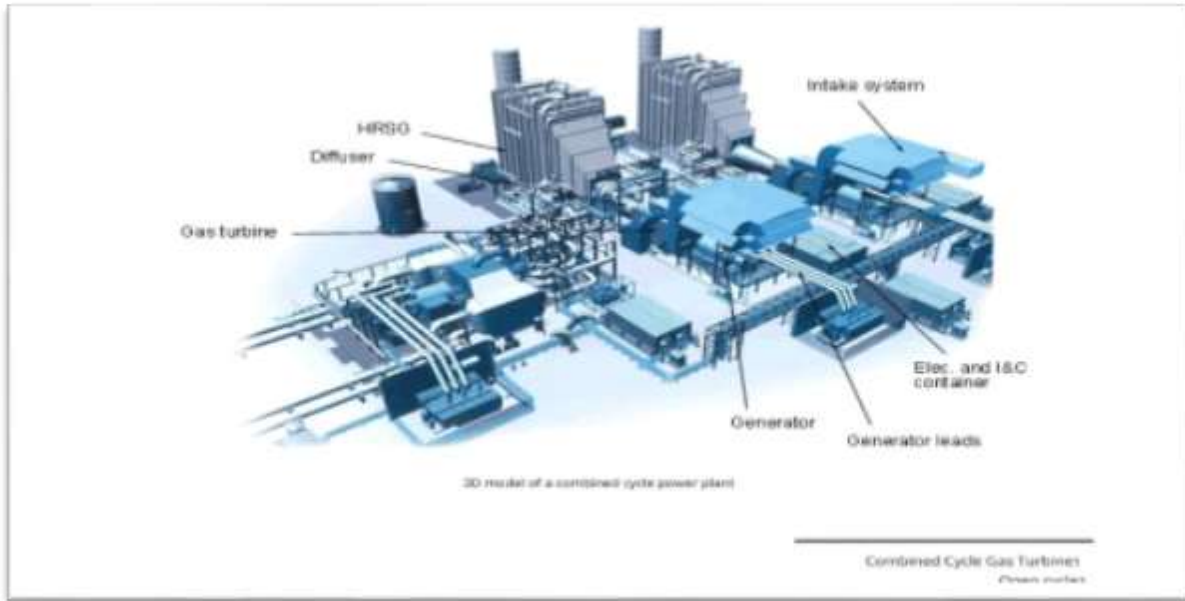


Figure 12 3D model of combined cycle

Working principles of the power plant CCGT the first step is the simple gas turbine power plant cycle. The open circuit gas turbine has a compressor, a combustion chamber and a turbine. For this type of cycle, the inlet temperature to the turbine is very high.

The output temperature of the resulting gases is also very high. The temperature is therefore high enough to provide heat for the second cycle, which operates using steam as the energy transfer fluid, as in thermal power plants. .[11]

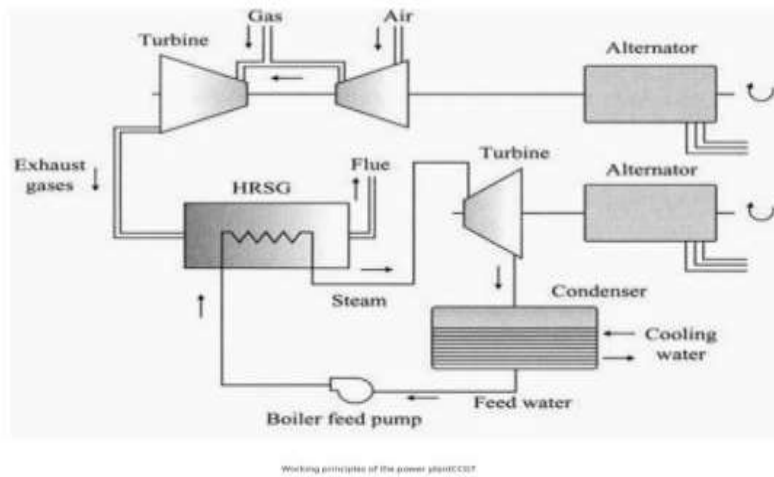


Figure 13 CCGT

Air is drawn through a large air intake section where it is cleaned, cooled and controlled. Heavy duty gas turbines, are able to work in a wide range of climatic and environmental conditions, due to the input of air purification systems that are specifically designed to suit the location of the power plant. Under normal conditions, the intake system is capable of processing air by removing contaminants to levels below levels harmful to the compressor and turbine. .[11]

In general, the incoming air has various pollutants. Which include:

In the gaseous form of pollution: Ammonia, Chlorine, Hydrocarbon gases, Sulfur in moldSO₂, and H₂S Discharge from the oil cooling holes

In the liquid form of pollution:

Chloride salt insoluble in water (sodium, potassium), Nitrate, Sulphate, Hydrocarbons

1.4.4 Plant Equipemnt

1.4.4.1 Turbine cycle

The air, which is pure, is compressed and mixed with natural gas and ignited, causing it to expand. The pressure from the expansion causes the turbine blades to rotate, which is about one axis¹ and a generator is connected, and electricity is created. In the second stage, heat from the chimney Gas turbine to produce steam by passing through a heat recovery steam generator (with temperature (HRSG). fresh steam Among 520 and 565 °C are used. .[11]

1.4.4.2 Heat recovery steam generator (HRSG)

In the heat recovery steam generator, very clean water flows in the pipes and hot gases pass around them and steam is produced as a result. The steam then turns the steam turbine and generator to generate electricity. Hot gases Leave the HRSG

at about 500°C and discharged into the atmosphere. Condensed steam and water system is similar to steam power plant. [11]

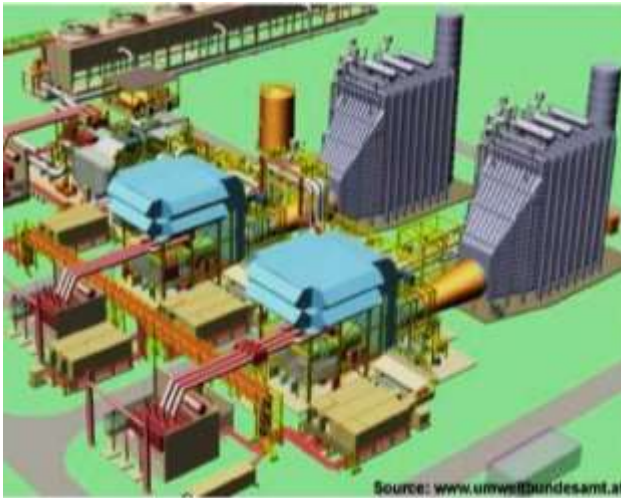


Figure .1.14. HRSG

1.4.5 Sample size and arrangement of power plants CCGT

The combined cycle system includes single-axis and multi-axis arrangement. The single axis system consists of a gas turbine, a steam turbine, a generator and a heat recovery steam generator. The combined cycle system includes single-axis and multi-axis arrangement. The single axis system consists of a gas turbine, a steam turbine, a generator and a heat recovery steam generator with gas turbine and steam turbine coupled to(HRSG) The generator is on a single axis.

Multi-axis systems have one or more gas turbine-generator and is the supply of steam through a main HRSG pipe common to a separate steam turbine-generator. In general investment terms, a multi-axis system about 5% has higher costs. [11] The main disadvantage of a multi-stage combined cycle power plant is that it increases the number of steam turbines, condensers and condensing systems, and perhaps cooling towers and circulating water systems to match the number of gas turbine.

1.4.6 Power plant efficiency CCGT

Approximately, the cycle of a steam turbine is one-third, and the cycle of an electric and gas turbine is two-thirds of the output power produce with CCGT (combined cycle power plant). The combination of both gas and steam cycles, high inlet temperature and low outlet temperature can be achieved. The efficiency of the cycles is also increased, since they are driven by the same fuel source. To increase the efficiency of the power system, it is necessary that be optimized, as an important bridge between gas turbine cycle and HRSG cycle The steam turbine is used to increase the output of the steam turbine. Function Great impact on the overall performance of the HRSG cycle power plant It has a combination. The electrical efficiency of a combined cycle power plant may be reach 56% and this is when the newly established power plant with output It has always been that these conditions are ideal. As we saw in single-cycle thermal units, combinedcycle units may also provide low-temperature thermal energy for industrial processes, central heating, and other applications, which is called cogeneration and is often referred to as a combined heat and power plant (CHP).It

is said (CHP). Efficiency It happens and is increased in HRSG gas turbine with afterburner and blade cooling. Supplementary combustion in CCPP Part of the compressed air flow is by passed and used to cool the turbine blades. It is necessary to use part of the energy of the chimney to compensate for the gas. This compensation can increase the efficiency of the power plant, especially when the gas turbine is operating under partial load. .[11]

1.4.7 Fuel for power plants CCPP

Turbines used in combined cycle power plants usually work with natural gas, which is more versatile than coal or oil and can 50% of energy applications should be used. Combined cycle power plants are usually powered by natural gas, although fuel oil, gas Synthetic or other fuels can be used.

1.4.7.1 Selective catalytic regeneration SCR

In order to control the emission of greenhouse gases in the flue gas so that it remains within the permissible level when it enters the atmosphere, the flue gas passes through two Catalyst located in passes HRSG.

One of the catalysts to control carbon monoxide (Ammonia is responsible for SCR. In addition to (NO_x), and other catalysts, nitrogen oxides (CO). Blue, (Ammonia blue is a mixture of (22% ammonia and 66% water) is injected into the system to further reduce NO_x levels. .[11]

1.4.7.2 Merits fuel efficiency

In conventional power plants, turbines have a fuel conversion efficiency of 33% have which means that two-thirds of the fuel for the turbine is off In the turbines of the combined cycle power plant, the fuel conversion efficiency is 55% or more, which means that they are about half the amount They burn the fuel to produce the same amount of electricity. [11]

1.4.7.3 Low capital costs

The capital cost of building a combined cycle unit is two-thirds of the capital cost of a coal-fired power plant. .[11]

1.4.7.4 Commercial availability

Combined cycle units are available from commercial suppliers anywhere in the world. They can be easily produced and transported. .[11]

1.4.7.5 Abundant fuel sources

The turbines used in combined cycle power plants run on natural gas, which is more flexible than coal or oil and can 50% of energy production to be To meet the energy demand, current power plants not only use natural gas but also other options such as biogas Derived from agricultural products work. .[11]

1.4.7.6 Less pollution and fuel consumption

Combined cycle power plants use less fuel in the production of each kilowatt/hour of electricity and produce less greenhouse gases than conventional thermal power plants, as a result of which the environmental damage caused by electricity production can be expected. Compared to the power plant that uses coal, the consumption of natural gas in the power plant CCPT is much cleaner.

1.4.7.7 Potential Applications In Developing Countries

The potential for combined cycle power plants is that some industries need electricity and heat or steam. For example, providing electricity and steam to a sugar refinery is of this type.

1.4.7.8 Harms

A gas turbine can only work with natural gas or high purity oil such as diesel fuel. Combined cycle power plants have economic justification in places where these types of fuels are available and are cheap. [11]

Chapter Two

Process Description

PROCESS DESCRIPTION

2.1 Introudction

The need for electricity usage is increasing with population growth, leading to an increase in the per capita energy consumption rate in general.

Companies, laboratories, and global universities have sought to provide alternatives and other types of energy, such as wind and solar energy. However, our research focuses on the potential development of power plants operating on gas or crude oil available in Iraq, The study included harnessing the thermal energy of combustion gases exiting gas turbine stacks to generate additional electrical power without the need for increased fuel combustion rates, leading to a reduction in thermal emissions and the rate of global warming, which has become a contemporary issue.

2.2 CCPP Process Description

The temperature of the combustion gases exiting the four gas turbines at one of the power generation stations in Basra ranges from 500 to 560 degrees Celsius, indicating high thermal energy that can be utilized to heat salt-free water and convert it into steam. Both gases and water enter the HRSG, with the water's inlet temperature being in two types: LP and HP, at 105°C and 106°C respectively, and the gas temperature at 519.9°C.

Heat exchange occurs within the HRSG to produce two types of steam, one at medium pressure and the other at high pressure. The gas temperature decreases from 520 to 120 degrees Celsius, which is very suitable compared to the ambient temperature, which does not exceed 55 degrees Celsius in Basra's summer season. Steam produced, both medium and high-pressure, is supplied to the steam turbines (two in number), with the steam exiting at a temperature of 60°C, generating

electrical power of 1844.5 MW, roughly equivalent to 50% of the gas turbines' energy.

Subsequently, the steam is cooled through the air cooler via heat exchange with the ambient air, reducing the steam temperature to 56.7°C, and the water is sent to a storage tank with replenishment from a makeup water tank.

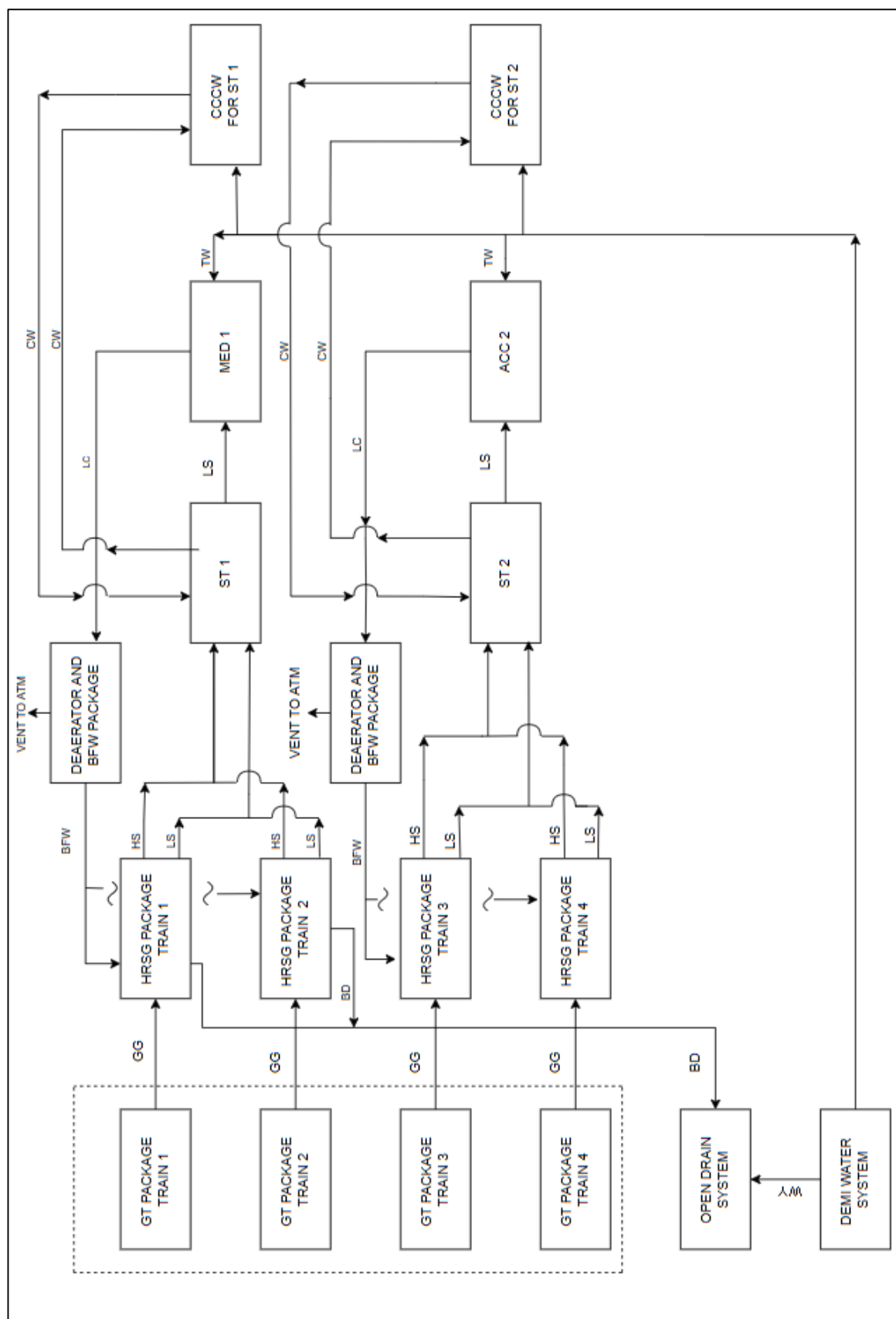
The water temperature in this tank decreases to the equilibrium temperature with the environment, ranging from 5 to 55 degrees Celsius annually, based on two theoretical calculations: the isothermal temperature of 15 and the temperature of maximum load of 55 degrees Celsius.

Then, the water is drawn through the pump to be circulated in the primary cooling system in the HRSG, with its temperature reaching approximately 21.275 degrees Celsius.

It is then sent through the same driving force of the pump to the deaerator, which works to reduce the dissolved oxygen level in the water to 200 ppm. Accompanying the oxygen isolation process is the supply of additional heat to the deaerator, which is provided by drawing a certain percentage of steam from the HRSG's outlet, as increased heat reduces the dissolved oxygen percentage in the water. The objective of reducing the oxygen percentage is to ensure that corrosion does not occur in the system. The water temperature is approximately 71.5 degrees Celsius.

Since the proposed HRSG operates with two temperature regimes, medium and low, and to ensure control over the flow rates for each part, it is suggested to install two pumps: the first one draws water from the deaerator and pumps it into the medium section of the HRSG, and the second one pumps water from the deaerator to the high section of the HRSG.

The process flow has been explained above from the moment the steam exits the HRSG until the liquid returns to the HRSG, and you can see the block diagram of it in fig [15].



*Figure 14 Combined Cycle
Power Plant Block Diagram*

Where:

FLUID TYPE	
CODE	NAME
BD	BLOW DOWN
BFW	BOILER FEED WATER
CW	COOLING WATER
GG	FLUE GAS
HS	HIGH PRESSURE STEAM
LC	LOW PRESSURE CONDENSATE
LS	LOW PRESSURE STEAM
TW	TREATED WATER (e.g. DEMI)
WY	DRAINAGES (CLEAR)

2.3 power plant located in the south of Iraq

And here is some power plant located in the south of Iraq:

PLANT NAME	LOCATION	OUTPUT
Rumaila shamara	Basrah	1460 MW
Shut AL-Basrah	Basrah	1250 MW
Khor Al-Zubair	Basrah	502 MW
Nasiriyah Thermal Power station	Nasiriyah	840 MW
Al Amarh	Amarh	500 MW
Dhi Gar combined cycle	Nasiriyah	750 MW
Haritha	Basrah	400 MW
Zubair Field power station	Basrah	740 MW
Samawa CCGT	samawa	750 MW
Hillah 2	Babylon	250 MW

Chapter Three

Literature Review

LITERATURE REVIEWS

3.1. 2023 Literature Reviews:

-In 2023, Nelson Ponce Junior, Jonas Rafael Gazoli, Alessandro Sete , Roberto M.G. Velasquez, Julian David Hunt, Fabio Tales Bindemann, Wilmar Wounnsoscky, Marcos Aurelio Vasconcelos de Freitas, Gabriela de Avila Condessa and Kamal Abdel Radi Ismail studied Climate impact on combined cycle thermoelectric power plants in hot and humid regions and **the results show that:**

- The wet bulb temperature is the meteorological factor with the greatest influence on the maximum generation of TPP Cuiaba. ´
- The higher the wet bulb temperature, the lower the maximum power achieved by gas turbines.
- The lower the power reached by the gas turbines, the lower the maximum power reached by the steam turbine.
- 6 °C of reduction in the wet bulb temperature represents an increase of 15.0 MW in the maximum power of the TPP, on average.
- 3.0 m/s of wind speed causes a 2.0 MW reduction in the maximum power of the gas turbines compared to days without wind.
- Wind speed also reduces the overall efficiency of TPP Cuiaba, as seen by the tendency for the plant's Heat Rate to increase.
- The STPF factor, created in this study to evaluate the ratio between the maximum powers of steam and gas turbines, tends to increase with wind speed.

- Due to the effects on the steam turbine power output, and the influence of meteorological conditions on the thermodynamic conditions, we propose that future work should deeply investigate the exhaust flue gas exiting the gas turbine.

-**In 2023**, Mrzljak Vedran, Baressi Šegota Sandi, Prpić-Oršić Jasna, and - Anđelić Nikola studied the Energy evaluation of a three-cylinder steam turbine that operates in combined cycle power plant and the most important results from this study were The most important **conclusions of this paper are:**

- Observing all the cylinders from the analyzed turbine, it is found that the dominant mechanical power producer is LPC, followed by the IPC, while HPC is the cylinder that produces the lowest mechanical power.
- Whole observed steam turbine from combined cycle power plant develop 119.41 MW of useful mechanical power.
- Energy loss and energy efficiency of all cylinders are reverse proportional – higher energy efficiency will result in lower energy loss and vice versa.
- IPC is the cylinder which has the lowest energy loss (equal to 2.59 MW) and the highest energy efficiency of 93.32%, while the highest energy loss (equal to 18.10 MW) and the lowest energy efficiency of 74.68% has LPC.
- Whole observed steam turbine from a combined cycle power the plant has energy loss equal to 23.43 MW, while its energy efficiency is equal to 83.60%. The observed steam turbine can be considered as a low-power steam turbine, so obtained energy loss and energy efficiency are in the expected range of such steam turbines.
- Steam mass flow rate through each cylinder is the main element which defines produced mechanical power and energy flows, while it did not have an obvious and direct impact on cylinder energy efficiency and energy loss.

-**On 20 July 2023**, Shiaoquo Chen, Ping Wang, David P. Hopkinson, Jared P. Ciferno, and Yuhua Duan studied the Analyses of hot/warm CO₂ removal processes for IGCC power plants and **the following conclusions** can be made from these analyses:

- A typical adsorption-based process will require a higher heat of adsorption when it is operated at a hot/warm temperature and accordingly, the electricity consumption of the process will also increase. Electricity loss for a hot/warm
- CO₂ capture process, for the most operating temperature range, is higher than the current baseline, the Selexol-based absorption process.
A desired process that may be advantageous is if the entire adsorption/desorption cycle is outside of the steam cycle. To be more precise, the adsorption temperature must be close to or above the highest steam temperature in a steam cycle.
- A novel potential sorbent should utilize the following reaction: Such a reaction will greatly reduce the entropy change of the reaction and thus significantly reduce its required heat of adsorption, which in turn will considerably improve the energy performance and thus the process economics.

3.2. (2022-2020) Literature Reviews:

-In 2022, Djamila Talah and Hamid Bentarzi studied the frequency Control System Effectiveness in a Combined Cycle Gas Turbine Plant ,Depending on the speed deviation characteristics and the frequency sensitivity to the load change, the action of the speed governor control results in a steady-state frequency deviation. The output power of the CCGT remains at more or less its rated value until the frequency of the system returns to its nominal value. As the frequency depends on the output power, the frequency should remain nearly constant in order to maintain the speed generator at its rated value.

Consequently, the reliability and effectiveness of the frequency control system is necessary for the stability of the power system

-In 2022, Andrey Rogalev, Nikolay Rogalev, Vladimir Kindra, Ivan Komarov and Olga Zlyvko studied the Development of Combined Cycle Power Plants Working on Supercritical Carbon Dioxide and **the result shows that;**

- The thermal scheme and mathematical model for the gas turbine combined cycle working on CO₂ have been developed. The optimal values of the key thermodynamic parameters have been identified for the case of gas turbine unit

GTE-160. It has been established, that at a temperature of 517 °C, the efficiency of the carbon dioxide Brayton cycle with recompression could be equal to 43.41% and achieve inlet and outlet turbine pressures of 24.0 and 8.5 MPa, respectively, and at the recompression percentage of 37.5%.

- The maximum net power generation of 55.1 MW is achieved with a zero recompression ratio with a net efficiency of 37.98% at a pressure of 30 MPa. This happens because the temperature of carbon dioxide at the inlet to the waste heat boiler decreases, therefore, the heat supply to the highly efficient cycle increases.
- When optimizing the secondary circuit, it was found that during the operation of the main Brayton cycle, the highest efficiency of the combined cycle is observed at pressures at the inlet and outlet of the turbine equal to 32 MPa and 8 MPa, at which the efficiency reaches 19.6%.
- Based on the mathematical simulation, it was found that replacing the conventional steam power plant, which operates in combination with GTE-160 gas turbine plant and uses carbon dioxide Brayton cycles, with recompression and base version provides a 1.2% increase in the net efficiency of the combined power plant. Such an increase in efficiency can be explained by a high average integral temperature of the heat supply in the Brayton cycle, the carbon dioxide temperature upstream of HE1 being about 311 °C, whereas, in the conventional design, this value is about 100 °C (the temperature downstream of the atmospheric deaerator)

-In 2021, Betty Puna, Ibrahim Alib , Raja Jadhava , Eric Chenc , Joe Selinger , Yue Zhang , and Gary Rochellec studied Advancing CO₂ Capture from Natural Gas Combined Cycle Power Plants with Piperazine Scrubbing the result shows that It is cost effective to supplement pilot plant projects to test low CO₂ flue gas conditions that are representative of NGCC. The pilot plant tests provided valuable reassurance for the operation of CO₂ absorption into aqueous PZ solvent, as well as confirmation that the effects of low inlet concentration do not impact regenerator performance. The ability of the Independence Model to predict process performance of low CO₂ flue gas is confirmed, building credibility for scaling up the aqueous PZ process for NGCC applications and integration. A few lessons learned for expanding a pilot project for multiple conditions: - Instrumentation may not be equally accurate for all conditions - Equipment sizing may be suboptimal; therefore, absolute performance could be impacted - Heat loss

is likely more significant when high temperature equipment is oversized - Long-term operation is needed to understand issues associated with degradation/corrosion and their mitigation - A wide range of operating conditions should be tested to evaluate models used in scale-up. Therefore, the goal for the NGCC pilot was never to demonstrate a certain target for % CO₂ absorption or a specific value of energy consumption, but to elucidate any specific issues associated with transferring the PZ technology to low CO₂ application and to generate relevant data for the evaluation of the modeling tool that would be used for scaling up and evaluating the solvent process for this application. While the tests in SRP and NCCC are a good first step for moving the PZ technology towards higher TRL for NGCC applications, neither site provided real NG flue gas. This is a worthwhile next step.

-In 2021, Vitaly Sergeev , Irina Anikina and Konstantin Kalmykov studied Using Heat Pumps to Improve the Efficiency of Combined-Cycle Gas Turbines

This paper studies the integration of heat pump units (HPUs) to enhance the thermal efficiency of a combined heat and power plant (CHPP). Different solutions of integrate the HPUs in a combined-cycle gas turbine (CCGT) plant, the CCGT-450, are analyzed based on simulations developed on “United Cycle” computer-aided design (CAD) system. The HPUs are used to explore low-potential heat sources (LPHSs) and heat make-up and return network water. The use of HPUs to regulate the gas turbine (GT) intake air temperature during the summer operation and the possibility of using a HPU to heat the GT intake air and replace anti-icing system (AIS), over the winter at high humidity conditions were also analyzed. The best solution was obtained for the winter operation mode replacing the AIS by a HPU. The simulation results indicated that this scheme can reduce the underproduction of electricity generation by the CCGT unit up to 14.87% and enhance the overall efficiency from 40.00% to 44.82%. Using a HPU with a 5.04 MW capacity can save \$309,640 per each MW per quarter.

The presented research work can be summarized in **the following points:**

1. Different solutions for integrating the HPUs in a combined-cycle gas turbine (CCGT) plant, the CCGT-450, are analyzed based on simulations developed on “United Cycle” computer-aided design (CAD) system.
2. Based on the results of the selection of low-potential heat sources and consumers, circuit solutions for the integration of HPUs were proposed:

- the use a HPU to heat the GT intake air and replace anti-icing system, over the winter at high humidity conditions;
 - the use of heat pumps for air cooling at the intake of a gas turbine unit in the summer period of operation;
 - the use of heat pumps for heating water of a heating unit (return network or make-up water).
3. The best solution was obtained for the winter operation mode replacing the anti-icing system by a HPU. This allows us to reduce the underproduction of electricity, associated with the losses because of the anti-icing system, up to 14.87%, when heating the inlet air from -2°C to 8°C . The overall efficiency of the CCGT unit increases from 40.00% to 44.82%. Using a HPU with a 5.04 MW capacity can save \$309,640 per each MW per quarter.

-In 2020, George Marin, Dmitrii Mendeleev, and Boris Osipov studied the operation of a 110 MW combined-cycle power unit at minimum loads when operating on the wholesale electricity market and **the result showed that:**

1. The dependences of the output parameters (high and low pressure steam flow rates, high and low pressure steam pressure,) on the loading of the combined cycle power unit were obtained. These dependencies make it possible to quickly determine the operating characteristics of the power unit.
2. The data obtained make it possible to predict changes in the main parameters of the combined cycle plant. [17-19]
3. Based on the research results, the constant minimum power of the CCGT unit was determined - 40 MW. (gas turbine capacity - 23.5 MW, steam turbine capacity - 16.5 MW). With such a minimum load, the power unit can operate for a long time.

-In 2020, Muchlisin¹, Nastopo Darmawan and Setya Budi studied The Effect of Decreasing Gas Turbine Flue Gas Temperature on The Performance of Muara Tawar Combined Cycle Power Plant.

Muara Tawar combined cycle power plant consists of three gas turbines, three heat recovery steam generators and one steam turbine. An update of gas turbine blades in 2016 effected on decreasing gas turbine flue gas temperature by 8°C up to 20°C. This study aims to analyse the effect of decreasing gas turbine flue gas temperature on output power of steam turbine. The output power of steam turbines is simulated by using Cycle-Tempo software, based on combined cycle heat balance. Variations of decreasing gas turbine flue gas temperature are configured on three gas turbines, two gas turbines and one gas turbine. The result shows that every 2°C decrease in flue gas temperature of three gas turbines will decrease output power of steam turbines by 0.787%. Whereas if decrease in flue gas temperature of two gas turbines and one gas turbine will decrease output power of steam turbine by 0.532% and 0.267% respectively. Meanwhile, the simulation results show that the decrease in the output power of the steam turbine after update of the gas turbine blades, 72.8% is caused by a decrease in gas turbine flue gas temperature and 27.2% by other factors.

3.3. (2019-2016) Literature Reviews:

-In 2019, Olesya Borush1, Pavel Shchinnikov, and Anna Zueva studied the Prospects of the Application of Dual-Fuel Combined Cycle Gas Turbine Units **and from the result**, we can see that:

- the efficiency values at a level of 55-56 % are achievable for parallel type CCGT with predominant solid fuel combustion.
- Dual-fuel parallel type CCGT is preferable to traditional pulverized coal power units in case the ratio of fuel prices of gas/coal does not exceed 5.
- Dual-fuel parallel type CCGT is preferable to binary discharge CCGT when the ratio of fuel prices gas/coal is more than 0,49-0,53. Thus, the development of dual-fuel combined cycle gas turbine unit technologies are a promising direction in the modernization and technical re-equipment of existing power plants in the coal regions.

-In 2019 ,D I Mendeleev, G E Maryin and A R Akhmetshin studied Improving the efficiency of combined-cycle plant by cooling incoming air using absorption refrigerating machine

and analyze the possibility of increasing the power of combined cycle gas turbine (CCGT) unit during the period of positive ambient temperatures. and studied usage of an absorption refrigerating machine in a CCGT cycle to increase its energy efficiency. The third aim is to analyze the operation of 110 MW combined-cycle power unit at various ambient temperatures, and to obtain alterations in the main CCGT characteristics when the ambient temperature changes. Calculations of the thermal scheme of a gas turbine were carried out using mathematical modeling, the steam turbine was calculated basing on the guidelines. The possibility of increasing the capacity of a CCP-110 MW was studied and analyzed. The conducted studies allowed them to conclude that the use of absorption refrigerating machine in the cycle of a combined-cycle plant can improve the efficiency of the unit, increase profits from power generation, and reduce penalties for noncompliance with the load schedule. and the conclusion is The use of an absorption refrigerating machine in the cycle of a combined-cycle gas turbine plant does not imply changes in the design of the main equipment (in particular, a gas turbine or compressor), thereby minimizing both costs and time for modernization The unit allows the equipment to operate at nominal (basic) parameters, which will result in removal of technical limitations on power and fulfillment of the load schedule, increasing output, profit, and will also reduce or completely eliminate penalties. Considering the fact that the equipment can be on optimal operating parameters for a longer time, its reliability, economy and safety are increased, and it is also possible to regulate power over wider limits (in the summer period), depending on the specified electrical load schedule . The total savings from the introduction of two ARMs for 2 GTU installations excluding operating costs will be 38,445,661 rubles

-In December 2019, Noor Kareem Naeem, Safaa Hameed Faisal, and Alaa Abdulrazaq Jassim wrote a Parametric Study of Combined and Cogeneration Systems: Energy and Exergy Approach This study reveals the effect of the operations conditions on the performance of the combined cycle and multi-stage flash unit in terms of energy and exergy.

- 1- The performance of CC depended on the operation conditions of the power plant. These conditions are pressure ratio, ambient temperature, and firing temperature.
- 2- The performance characteristics of MSF which is (the thermal performance ratio (PR)) depends on the pressure ratio, ambient temperature, and firing

temperature and could vary only with the ratio of distilled flow rate to the heating steam.

-In 2019 ,Dan- teodor Bălănescua and Vlad- Mario Homutescua studied Performance analysis of a gas turbine combined cycle power plant with waste heat recovery in Organic Rankine

Gas-steam combined cycle power systems currently are the most advanced power systems operating with fossil fuels since offer the highest efficiency level and lowest pollution. Even so, amount of waste heat released with flue gas into the atmosphere is still significant. A typical way to recover it is heat production but this solution is not viable when there is no heat demand. The interest in this case is for production of additional power. A possible solution is to add an organic Rankine cycle unit downstream the gas-steam combined cycle power plant, which thus is converted into a gas-steam-organic combined cycle power plant. The performance of such a power system, based on Orenda OGT15000 gas turbine with 16.5 MW output in simple cycle, is analyzed in the paper. Besides, the fuel savings and their cost are assessed. Two organic working fluid were considered, namely R134a and R123. The study shows that efficiency of the power plant increases by roughly 1.1 % when the ORC unit is added. Taking into account the current concerns regarding the fossil fuel depletion, the estimated fuel savings could be considered significant

-On January 16, 2017, Asad Dehghani Samani studied the Combined cycle power plant with indirect dry cooling tower forecasting using an artificial neural network. In this study, the steam turbine of a combined cycle power plant with the dry cooling tower was modeled using MLP with backpropagation training. First of all, the main cooling system was modeled to predict the cooling capacity in the steam turbine exhaust using the data available to the operator. In this manner, the operators can predict the exhaust steam vacuum of the ST, which is critical in the ST output, with good accuracy. Then the data was used to predict the power output of the ST using data available to the operators through the power plant's data warehouse. It can be seen that ANN modeling is capable of predicting the electrical production of ST under varying load conditions of two gas turbines. The prediction can be utilized in two ways. First, it can be integrated into a dynamic condition monitoring system in which the online performance is compared to the derived model and any deviations are diagnosed and inspected.

This can ensure safe and reliable operation in various conditions. Second, the model can be used for accurate power production forecasts. These forecasts are used in the electrical power market.

-In 2016, Mahdi Sharifzadeh and Nilay studied the scale-up and integration of a solvent-based carbon capture process into a natural gas combined cycle (NGCC) power plant for a novel solvent, GCCmax, and the MEA reference solvent. The aim was to establish and quantify the superior performance of the new solvent at an industrial scale. Furthermore, it provided in-depth insights into the retrofit and flexible operation of NGCC power plants. It was observed that the control strategy for the combined cycle gas turbine (CCGT) during load reduction, has profound implications for the flow rate and composition of flue gas, and hence affects carbon capture costs. It was also observed that NGCC power plants are less efficient in part-load operational scenarios. In the present research, the method of integrated process design and control was adapted and solved. The proposed optimization algorithm successfully established a tradeoff between the design and operational criteria. The overall total annual costs in terms of capital investment and energy costs were minimized while the process operability was ensured under all load reduction scenarios.

3.4. (2013-2011) Literature Reviews:

-In 2013, Arvind Kumar Tiwari, Mohd. Muzaffarul Hasan and Mohd. Islam Studied the Effect Of Ambient Temperature On The Performance Of A Combined Cycle Power Plant, and the result shows that the following conclusions can be made by varying the ambient temperature:

1. The combined cycle loses its efficiency by about 0.04% for every °C rise in ambient temperature.
2. The gas turbine cycle efficiency decreases by 0.03 to 0.07% for every °C rise in ambient temperature.
3. The exergy destruction in the combustion chamber decreases with an increase in ambient temperature from 0.21 to 0.34% for every °C rise in ambient temperature.

4. The exergy destruction in compressor, gas turbine, HRSG, and steam turbine increases with an increase in ambient temperature from 0.32 to 0.35% for every °C rise in ambient temperature.
5. The ambient temperature also affects the exergy loss via exhaust by 0.92 to 1.14% for every °C rise in ambient temperature.
6. The air-fuel ratio increases with increase in ambient temperature

-On 2013, Ibrahim Thamir K. and Rahman M.M studied the effective parameter of the triple-pressure reheat combined cycle performance, and The simulated modeling results show that the influence of ambient temperature, compression ratio, and TIT significantly affects the overall performance of the CCGT power plant. The simulated modeling results are as follows. The compression ratios and TIT are strongly influence on the overall efficiency of the combined cycle GT power plant. The increase in the ambient temperature has a significant influence on this type of power plant. The power output and the overall efficiency of a combined cycle GT power plant decrease with an increase in the ambient temperature. Higher overall efficiency for the CCGT power plant was found at about 59% with TIT 1900 K. The overall thermal efficiency increases and total power output decreases linearly with the increase in the compression ratio with constant TIT

-In 2012, Thamir K. Ibrahim and M. M. Rahman studied the Effect of Compression Ratio on the Performance of Combined Cycle Gas Turbine and the result shows that The complete model for the combined-cycle gas-turbine power plant with the effect of the gas-turbine peak compression ratio has been used for carrying out the thermodynamic study. The simulated modeling results show as follows:

- The compression ratios, air-to-fuel ratio as well and the isentropic efficiencies are strongly influenced by the overall thermal efficiency of the combined-cycle gas-turbine power plant.
- Higher overall efficiencies for combined-cycle gas turbines compared to gas-turbine plants. The efficiency quoted ranges from about 61%.
- The overall thermal efficiency increases and total power output decreases linearly with the increase of the compression ratio with constant turbine inlet temperature.

- The peak overall efficiency occurred at the higher compression ratio with the higher cycle peak temperature ratio as well as higher isentropic compressor and turbine efficiencies

-On February 2011, Keyvan Daneshvar, Ali Behbahani-nia, Yasamin Khazraii, and Azam Ghaedi studied the transient Modeling of Single-Pressure Combined Cycle Power Plant Exposed to Load Reduction y. This study considered load reduction in gas turbine as a dynamic behavior of cycle and its result have been shown in some figures. The model output at the end of transient operation (during which there was no change in parameter values and system was approximately in steady-state mode of operation) was compared with the numerical output of well-known commercial software. The agreement between these two groups of data was good.

-In 2011, Thamir K. Ibrahim, M. M. Rahman, and Ahmed N Abdalla studied the Optimum Gas Turbine Configuration for Improving the performance of a Combined Cycle Power Plant and the result shows the combined cycle with a topping cycle: simple gas turbine, two shaft gas turbine, intercooler gas turbine, and regenerative gas turbine. The simulated modeling results show that the influence of the ambient temperature and compression ratio are significantly effect on performance of combined cycle gas turbine power plant for different gas turbine configuration. The results are summarized as follows: 1. The ambient temperature and compression ratios are strongly influence on the overall thermal efficiency of the combined cycle gas turbine power plant for different gas turbine configuration. 2. Higher overall thermal efficiency for combined cycle gas turbine with regenerative gas turbine configuration. Efficiency quoted range about 64.5% with a low compression ratio. 3. The overall thermal efficiency of combined cycle decreases and total power output increases linearly with increase of ambient temperature for all gas turbine configurations except the regenerative gas turbine the total power output decreases.

-On September 2011 Experimental Studies on Gas Turbine Engine Dynamics By Tengku Norhanisah Bt Tengku Kamaruddin. The objectives of this study are to study the concept of rotor dynamics and develop rotor dynamic experimental studies based on previous journals. Result .. In conclusion, the study of gas turbine rotodynamics using experimental study is one of the possible solutions to understanding the behavior of gas turbines. In this study, the design data of gas

turbines are not available. However, the analysis is done using an experimental study. Based on the result, the shaft is found not to be at the center. The shaft is positioned more to the left at the point of the compressor. Besides that, based on peak-to-peak analysis, the amplitude of the gas turbine is proportional to the speed of the gas turbine. The final result from this project is acceptable with few recommendations for future analysis. It is proven that the three objectives of this project are achieved.

-On September 2011, Christoph Ruchti, Hamid Olia, Karsten Franitza, Andreas Ehram, and Wesley Bauver studied the Combined Cycle Power Plants as the ideal solution to balance grid fluctuations.

The result shows that The time for a hot start could be reduced to less than 30 minutes without significant investment in additional or increased equipment. A key requirement for success is the close cooperation of all equipment designers in a true Plant Integrator spirit. Detailed computation methods for stress calculation of critical components have been openly shared and implemented in the dynamic simulation of the overall behavior. This made it possible to assess the benefit of operation concept modifications on each simulation run, without the need to request a lifetime assessment report from the component designer.

Further improvement in plant flexibility may be expected through such a collaborative approach.

-In 2011, Abigail González-Díaz, Agustín M. Alcaráz-Calderón, M.O.

González-Díaz, Ángel Méndez-Aranda, Mathieu Lucquiaud, and Jose Miguel González-Santaló studied the effect of the ambient conditions on gas turbine combined cycle power plants with post-combustion CO₂ capture and A comprehensive assessment shows the operation of NGCC plants with CO₂ capture is resilient to changes in ambient conditions, such as temperature, pressure and humidity. The results of the models developed in this study are in good agreement with available data in the literature, as shown in table 7 and Figures 14 to 16. Although atmospheric pressure is an important parameter to define the generating capacity and to design a gas turbine, it has no impact on efficiency once the plant is installed. Nonetheless, atmospheric pressure does have a significant effect on power output, although this is not caused by the capture plant. Similarly, the effect of relative humidity is not found to be significant. The ambient temperature is an important variable that affects both the power generating capacity and the efficiency of the power plant, due to the effect on the

air mass flow and pressure ratio of the gas turbine. The power decreases from 700 MW to 530 MW and the efficiency from 50.8% to 48% when the temperature increases to 45 °C for a plant designed for ISO conditions at 15°C. The ambient temperature also has an impact on the levelised cost of electricity from 52.58 \$/MWh to 64 \$/MWh when the ambient temperature increases from -5 °C to 45°C. In the design of a NGCC plant with capture, the operating practices of the electricity market must be taken into consideration. It is proposed that the design include a low pressure steam turbine capable of operation with the capture plant by-passed, but that the condenser be designed when the capture plant is in operation. This approach gives the flexibility of operating the NGCC at full power even if the capture plant is off line, and allows the operation of the NGCC without the capture plant with a small penalty in efficiency due to a higher condenser pressure. An understanding of the performance of NGCC plants with CO₂ capture provides a good basis for defining relevant operating procedures. A flexible design also allows trade-offs between capture levels and electricity output by adjusting the carbon intensity of electricity generation under severe ambient conditions.

3.4. (2008-2002) Literature Reviews:

-On February 5, 2008 , Hiwa Khaledi Roozbeh Zomorodian and Mohammad Bagher Ghofrani studied the effect of Inlet Air Cooling by Absorption Chiller on Gas Turbine and Combined Cycle Performance

as turbine performances are directly related to site conditions. The use of gas turbines in combined gas-steam power plants, also applied to cogeneration, increases such dependence. In recent years, inlet air cooling systems have been introduced to control air temperature at compressor inlet, resulting in an increase in plant power and efficiency. In this paper, the dependence of outside conditions for a simple gas turbine and a combined cycle plant is studied, using absorption chiller as inlet air cooling system.

as case study, a simple plant equipped with one frame E gas turbine and a combined cycle with a two pressure level heat recovery steam generator (HRSG). It was found that inlet air cooling with absorption chiller has great positive influence on power and less on efficiency of the gas turbine plant. Two steam sources (External and Internal) have been considered for chiller. External source has large positive influence on power but keep the efficiency of the combined

cycle unchanged, while internal source causes a reduction in steam turbine mass flow. Consequently power production and efficiency of the combined cycle.

-In 2008, Masao Ishikawa, Masashi Terauchi, Toyoaki Komori, And Jun Yasuraoka studied the Development of a High-Efficiency Gas Turbine Combined Cycle Power Plant and the results showed that the Gas turbine combined cycle power plants mainly using MHI D, F and G-type gas turbines are currently in operation in large numbers, and there are many under construction or planned both in Japan and abroad. The development and introduction of this new technology respond to the need for the effective utilization of energy and countermeasures against global warming, thus contributing greatly to society. MHI is determined to continue efforts to develop new technology as a pioneer company in this field.

-In 2007, Tadashi Gengo, Yoshinori Kobayashi, Nagao Hisatome, Tatsuo Kabata, Yoshimasa Ando, and Kenichiro Kosaka studied the Development of HighEfficiency SOFC Combined Cycle System and from the result we can see that the High-efficiency power generation as a special feature of SOFC can be best utilized in a combined SOFC gas turbine-steam turbine power generation system. High efficiency power generation will be 70% LHV or above when natural gas is used and 60% or above when coal gas is used. We expect to develop the technologies needed for both system development and elemental development of the SOFC micro gas turbine combined cycle system, thus contributing, by marketing the product, to a society using environmentally-friendly energy in the 21st century.

-In 2007, Charles W. Forsberg and James C. Conklin studied the Nuclear-Fossil Combined-Cycle Power Plant for Base-Load and Peak Electricity and the result shows that the Combined nuclear-fossil systems have the potential to offer the best of both worlds: low-cost base-load electricity and lower-cost peak power relative to the existing combination of base-load nuclear plants and separate fossil-fired peakelectricity production units. Significant work is required to fully understand and develop this new electricity generation option.

In January 2005, Paolo Chiesa, Giovanni Lozza, and Luigi Mazzocchi studied the possibility to burn hydrogen in a large-size, heavy-duty gas turbine designed to run on natural gas as a possible short-term measure to reduce greenhouse

emissions of the power industry "Using Hydrogen as Gas Turbine Fuel" and The simulations carried out in this work allow a positive answer to the issues bines. However, an SFT -related to hydrogen combustion in modern gas tur

abatement to about 2300 K seems necessary to comply with NOx emission limits without incurring excessive

operating costs of the end-of-pipe de-nitrification systems This is possible without dramatic performance losses by means of a massive fuel dilution with steam or nitrogen ~the latter providing -minor losses of efficiency!. Different strategies have been envis aged to operate the gas turbine in the presence of dilution. Looking at the VGV operated solution ~which appears the most likely for the first realizations! the efficiency loss is limited to 0.9 points for nitrogen dilution and 1.9 for steam dilution. Equally small is the influence on the combined cycle power output provided that the gas turbine power output can be increased ~by about 10%! in -consequence of the compressor airflow reduction. The other solu tions here investigated ~increased pressure ratio and re-engineered machine! are not particularly attractive in terms of efficiency but provide a much larger power output, an opportunity to reduce the specific costs provided that engineering costs are divided upon a -sufficient number of units. It must also be noticed that VGV op erations reduce the part-load capabilities of the machine, but make the gas turbine rather insensitive to elevated ambient temperatures the “natural” power loss can be compensated by re-opening the~

VGV’s!. As a final consideration on system costs, it can be said that steam dilution allows for reduced capital cost compared to nitrogen, even if providing lower efficiency. In fact, a smaller steam turbine and condenser can be adopted, while the N2 dilution requires a bulky and expensive additional compressor.

-On May 7, 2004, Lauren Tsai studied the Design and Performance of a turbine engine from an Automobile Turbocharger. The purpose of this project was to design and fabricate a gas turbine engine for demonstration use in these two classes. The engine was built from an automobile turbocharger with a combustion chamber connected between its compressor and turbine. Pressure and temperature sensors at different points of the engine cycle allow students to monitor the performance of the individual engine components and the complete engine cycle. The purpose of this project was to design and fabricate a gas turbine engine that is suitable for use as a classroom demonstration. The gas turbine designed and built for this project illustrates the operation of the Brayton cycle. The Brayton cycle is a convenient cycle to demonstrate because it involves a combination of standard components, which are used in many other energy conversion applications. The Brayton cycle consists of a compressor, a heat exchanger, a

turbine, and another heat exchanger. The gas turbine engine operates on an open version of the Brayton cycle and allows students to measure the temperature and pressure changes associated with each system component. The students can then use these values to calculate the performance of the engine components as well as the overall cycle efficiency.

- **In 2002**, JACEK TOPOLSKI and JANUSZ BADUR studied the Comparison of the combined cycle efficiencies with different heat recovery steam generators and the result shows that the Traditional HRSG design is essentially based on manufacturer experience and heuristics to obtain convenient matching of temperature drop, and pressure drop and the exchange surface area. Taking advantage of the fact that HRSG is thermodynamically determined by the knowledge of working fluid temperature and its enthalpies, it is possible, using a computer code like the COM-GAS, to find a more optimal and consistent exchanger arrangement. The present study which is some kind of optimization analysis has shown that in the case of 3PA, the exchanger's arrangement is not too far from optimum.- L.O. Tomlinson and S. McCullough studied the SingleShaft Combined-Cycle Power Generation System and the result shows that Combined-cycle systems provide reliable and economic service in electric utility power generation applications. Flexibility in equipment selection and arrangement, thermal cycle, type of fuel, emission control, and duty cycle enable optimization to meet a wide variety of owner requirements. Single-shaft combined-cycle systems offer further advantages as follows:

- Simplicity of single-unit control
- Low plant cost
- Minimum land area use
- Simplified control and operation with multiple pressure and reheat steam cycles
- High reliability and availability

Therefore, consideration of the single-shaft unit is recommended for all power generation combined cycles that are installed in a single phase.

-**On May 2002**,y G.H. Abd-Alla studied the using exhaust gas recirculation in internal combustion engines The aim of this work is to review the potential of exhaust gas recirculation (EGR) to reduce the exhaust emissions, particularly NO_x emissions, and to delimit the application range of this technique.

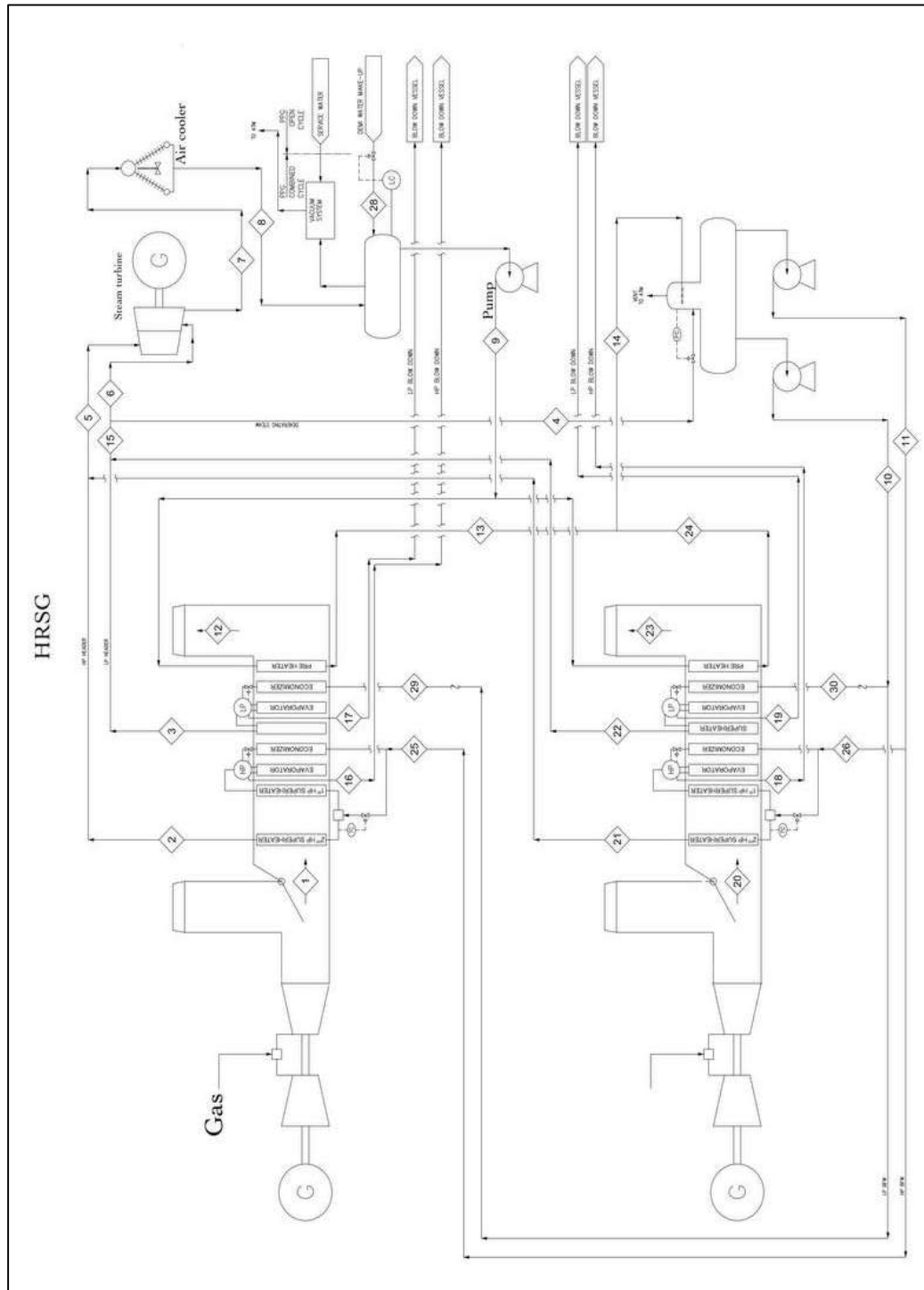
A detailed analysis of previous and current results of EGR effects on the emissions and performance of Diesel engines, spark ignition engines and dual fuel engines is introduced. From the deep analysis, it was found that adding EGR to the air flow rate to the Diesel engine, rather than displacing some of the inlet air, appears to be a more beneficial way of utilizing EGR in Diesel engines. This way may allow exhaust NO_x emissions to be reduced substantially.

In spark ignition engines, substantial reductions in NO concentrations are achieved with 10% to 25% EGR. However, EGR also reduces the combustion rate, which makes stable combustion more difficult to achieve. At constant burn duration and brake mean effective pressure, the brake specific fuel consumption decreases with increasing EGR. The improvement in fuel consumption with increasing EGR is due to three factors: firstly, reduced pumping work; secondly, reduced heat loss to the cylinder walls; and thirdly, a reduction in the degree of dissociation in the high temperature burned gases. In dual fuel engines, with hot EGR, the thermal efficiency is improved due to increased intake charge temperatures and reburning of the unburned fuel in the recirculated gas. Simultaneously, NO_x is reduced and smoke is reduced to almost zero at high natural gas fractions. Cooled EGR gives lower thermal efficiency than hot EGR but makes possible lower NO_x emissions. The use of EGR is, therefore, believed to be most effective in improving exhaust emissions.

Chapter Four

Material & Energy Balance

COMBINED CYCLE POWER PLANT MATERIAL AND ENERGY BALANCE



COMBINED CYCLE FLOW DIGRAM

4.1 Case 1

- The ambient temperature=55 C
- The ambient pressure= 1.007 bar
- The ambient relative humidity= 40%

4.1.1 HRSG Material And Energy Balance

4.1.1.1 HRSG1 And HRSG2 Material Balance:

.The flow rates of NO_x and CO emissions based on their concentration limits:-

- NO_x emission limit: <114 mg/m³
- CO emission limit: <450 mg/m³

Since we don't have the exact concentrations, we'll calculate the maximum possible flow rates based on the concentration limits:

Flow rate of NO_x = (114 mg/m³ / 1000000 mg/kg) * (1941500 kg/h / 0.45 kg/m³) = 491.846 kg/h

Flow rate of CO = (450 mg/m³ / 1000000 mg/kg) * (1941500 kg/h / 0.45 kg/m³) = 1941.5 kg/h

.N₂, CO₂, and H₂O Gases flow rate:-

Total flow rate of NO_x and CO = 491.846 kg/h + 1941.5 kg/h = 2433.346 kg/h

Gases flow rate = Total mass flow rate - Flow rate of NO_x and CO
.654 kg/h = 1939066kg/h - 1094.724 kg/h 1941500

4.1.1.2 HRSG1 And HRSG2 Energy Balance:

we will use the energy balance equation for each heat exchanger in the HRSG,
The energy balance equation is:

$$Q = \dot{m}_g c_{p,g} (T_{g,in} - T_{g,out}) = \dot{m}_s (h_{s,out} - h_{s,in})$$

where Q is the heat transfer rate, $\dot{m}$ is the mass flow rate, cp is the specific heat capacity, T is the temperature, h is the enthalpy, and the subscripts g and s denote the gas and steam streams, respectively.

- For the LP stream, the inlet enthalpy is still given as $h_{s,in}=441.464$ kJ/kg. The outlet enthalpy can be found from the steam tables using the outlet pressure of $P_{s,out}=7.14$ bara. Assuming the steam is superheated at $T_{s,out}=220$ °C, the outlet enthalpy is $h_{s,out}=2798.6$ kJ/kg. The heat transfer rate for the LP stream is then:

$$Q_{LP} = m_{s,LP} (h_{s,out} - h_{s,in})$$

$$= 20.3(2798.6 - 441.464) = 47777.4 \text{ kW}$$

- The gas outlet temperature after the LP heat exchanger can be found by rearranging the energy balance equation:

$$T_{g,out,LP} = T_{g,in} - m_g c_{p,g} Q_{LP}$$

$$= 519.9 - 539.3 \times 1.005 \times 47777.4 = 431.4^\circ\text{C}$$

- For the HP stream, the inlet enthalpy is still given as $h_{s,in}=454.238$ kJ/kg. The outlet enthalpy can be found from the steam tables using the outlet pressure of $P_{s,out}=83.63$ bara. Assuming the steam is superheated at $T_{s,out}=502$ °C, the outlet enthalpy is $h_{s,out}=3499.4$ kJ/kg. The heat transfer rate for the HP stream is then:

$$Q_{HP} = m'_{s,HP} (h_{s,out} - h_{s,in})$$

$$= 59.6(3499.4 - 454.238) = 181569.5 \text{ kW}$$

- The gas outlet temperature after the HP heat exchanger can be found by rearranging the energy balance equation:

$$T_{g,out,HP} = T_{g,out,LP} - m'_g c_{p,g} Q_{HP}$$

$$= 431.4 - 539.3 \times 1.005 \times 181569.5 = 103.8^\circ\text{C}$$

- This is the same as the gas outlet temperature, confirming the energy balance.
- Therefore, the outlet temperatures of the LP and HP streams are:

$$T_{s,out,LP} = 220^\circ\text{C}$$

$$T_{s,out,HP} = 502^\circ\text{C}$$

4.1.2 Steam Turbine Material And Energy Balance:

we need to use the following equations:

- Material balance:
 - $m_{in} = m_{out}$
- Energy balance:
 - $m_{in} h_{in} = m_{out} h_{out} + W$
- Bernoulli equation:
 - $p_{in} + \frac{1}{2} * \rho_{in} v_{in}^2 = p_{out} + \frac{1}{2} * \rho_{out} v_{out}^2$
- Efficiency:
 - $\eta = (h_{in} - h_{out}) / (h_{in} - h_{out,s})$

Using the Steam table, we can find the specific enthalpy and density of the inlet and outlet streams:

Inlet stream:

$$H_{in} = 0.5 \times (118.5 \times 3435.9 + 37.6 \times 2793.7) = 228.8 \times 10^3 \frac{\text{kJ}}{\text{kg}}$$
$$\rho_{in} = 0.5 \times (118.5 \times 4.7 + 37.6 \times 19.8) = 5.9 \text{ kg/m}^3$$

Outlet stream:

$$h_{out,s} = 1915.0 \text{ kJ/kg}, \quad \rho_{out} = 0.6 \text{ kg/m}^3$$

Using the isentropic efficiency, we can find the actual outlet enthalpy:

$$h_{out} = h_{in} - \eta \times (h_{in} - h_{out,s})$$
$$= 228.8 \times 10^3 - 0.5 \times (228.8 \times 10^3 - 1915.0) = 2108.5 \text{ kJ/kg}$$

Using the energy balance, we can find the work output:

$$W = m_{in} h_{in} - m_{out} h_{out}$$
$$= 156 \times (228.8 \times 10^3 - 2108.5) = 1844 \text{ MW}$$

Using the Bernoulli equation, we can find the outlet velocity:

$$V_{out} = 130.4 \text{ m/s}$$

Using the Steam table, we can find the outlet temperature:

$$\text{Outlet temperature: } T_{out} = 60.1 \text{ }^\circ\text{C}$$

Therefore, the work output of the steam turbine is **1844.57 MW** and the outlet temperature is **60.1 °C**.

4.1.3 Air Cooler Condenser Material And Energy Balance:

To find the outlet temperature of the steam ($T_{out,s}$), we need to apply the energy balance equation for the air cooler condenser:

$$\dot{m}_s(h_{in,s} - h_{out,s}) = \dot{m}_a(cp)_a(T_{out,a} - T_{in,a})$$

where $\dot{m}_s$ is the mass flow rate of the steam, $h_{in,s}$ and $h_{out,s}$ are the specific enthalpies of the inlet and outlet steam, $\dot{m}_a$ is the mass flow rate of the air, $(cp)_a$ is the specific heat capacity of the air, and $T_{in,a}$ and $T_{out,a}$ are the inlet and outlet temperatures of the air.

We are given the following values:

- $\dot{m}_a = 156 \text{ kg/s}$
- $T_{in,s} = 60.1 \text{ °C}$
- $P_{in,s} = 0.24 \text{ bara}$
- $P_{out,s} = 0.56 \text{ bara}$
- $T_{in,a} = 25 \text{ °C}$

using SteamTable, we get:

- $H_{in,s} = 2675.9 \text{ kJ/kg}$
- $H_{out,s} = 2643.4 \text{ kJ/kg}$

To find $\dot{m}_a$, we can use the fan performance curve or the manufacturer's data. For example, assuming a fan efficiency of 80% and a power consumption of 100 kW, we can use the formula:

$$\dot{m}_a = P_{fan}\eta_{fan}/\Delta p_a$$

where P_{fan} is the fan power, η_{fan} is the fan efficiency, and Δp_a is the pressure difference across the fan. Assuming the inlet and outlet pressures of the air are equal to the atmospheric pressure of 1.013 bar, we get:

$$\dot{m}_a = 101300100 \times 10^{-3} \times 0.8 = 0.789 \text{ kg/s}$$

To find $(cp)_a$, we can use the specific heat of air at constant pressure, which depends on the temperature and pressure of the air, the specific heat of air at 25 °C and 1.013 bar is about 1.006 kJ/kg K. Therefore, we can use this value as an approximation for $(cp)_a$.

To find $T_{out,a}$, we can use the energy balance equation and solve for $T_{out,a}$:

$$T_{out,a} = T_{in,a} + (m'_s(h_{in,s} - h_{out,s})/m'_a(cp)_a$$

Plugging in the values, we get:

$$T_{out,a} = 25 + 156 \times (2675.9 - 2643.4)/0.789 \times 1.006 = 25 + 64.6 = 89.6 \text{ } ^\circ\text{C}$$

Finally, to find $T_{out,s}$, we can use the energy balance equation and solve for $T_{out,s}$:

$$T_{out,s} = T_{in,s} - m'_a(cp)_a(T_{out,a} - T_{in,a})$$

Plugging in the values, we get:

$$T_{out,s} = 60.1 - 0.789 \times 1.006 \times (89.6 - 25)/156 = 60.1 - 3.4 = 56.7 \text{ } ^\circ\text{C}$$

4.1.4 Deaerator Material And Energy Balance:

Given:

- $m_{(water)} = 156 \text{ kg/s}$
- $h_{\{steam\}} = 2881.68 \text{ kJ/kg}$
- $h_{(water)} = 251.18 \text{ kJ/kg}$
- $m_{\{steam\}} = 2.8 \text{ kg/s}$ (it comes from 7% of the outlet streams from the HRSG1 and HRSG2)

The energy balance equation is:

$$(m_{\text{steam}} * h_{\text{steam}}) + (m_{\text{water}} * h_{\text{water}}) - (m_{\text{steam}} + m_{\text{water}}) * h_{\text{outlet}}$$

Plugging in the values:

$$(2.8 \text{ kg/s} * 2881.68 \text{ kJ/kg}) + (156 \text{ kg/s} * 251.18 \text{ kJ/kg}) = (2.8 \text{ kg/s} + 156 \text{ kg/s}) * h_{\text{outlet}}$$

Solving for h outlet:

$$H_{\text{outlet}} = [(2.8 * 2881.68) + (156 * 251.18)] / 2.8 + 156$$

$$H_{\text{outlet}} = [(8078.704) + (39184.08)] / 158.8$$

$$H_{\text{outlet}} = 47262.784 / 158.8$$

$$H_{\text{outlet}} = 297.57 \text{ kJ/kg}$$

The specific enthalpy of the outlet water from the deaerator is 297.57 kJ/kg. According to the Engineering Toolbox, the temperature corresponding to a specific enthalpy of approximately 297 kJ/kg for water is close to 105°C.

4.1.5 Pumps Material And Energy Balance:

4.1.5.1 The First Single Pump:

To calculate the work of the pump using the Bernoulli equation, we will use the following formula:

$$W = \Delta H + \frac{p_2 - p_1}{\rho} + g \cdot \Delta z + \frac{u_2^2 - u_1^2}{2}$$

Where

- W = work done by the pump
 - $\Delta H = h_2 - h_1$ = change in enthalpy
 - p_1, p_2 = initial and final pressure
 - ρ = density of the fluid
 - u_1, u_2 = initial and final velocities
- $P_1 = 0.1707 * 10^5 \text{ pa}$
 - $P_2 = 59.676 * 10^5 \text{ pa}$

- $U_1 = 1.5 \text{ m/s}$

To calculate the velocity out side use the principle of continous formula

First calculate the area at both points

$$A = \pi r^2$$

$$A_1 = 25\pi, A_2 = 36\pi$$

$$U_2 = \frac{U_1 A_1}{A_2} = \frac{1.5 \cdot 25\pi}{36\pi}$$

- $U_2 = 1.04 \text{ m/s}$
- $T_1 = 56.7 \text{ C}$
- $T_m = 275.22 \text{ C}$
- from steam table
- $H_1 = 384.97 \text{ kJ/kg}$
- $H_2 = 1212.0 \text{ kJ/kg}$
- $P_1 = 0.1707 \cdot 10^5 \text{ pa}$
- $P_2 = 0.977 \cdot 10^5 \text{ pa}$

Now let calculate the change in enthalpy

$$\Delta H = h_2 - h_1$$

$$384.97 \text{ kJ/kg} - 1212.0 \text{ kJ/kg} = -827.03 \text{ kJ/kg}$$

Now let calculate the change in elevation

$$\Delta Z = z_d - z_s = 10 - 0 = 10 \text{ m}$$

$$W = \Delta H + \frac{p_2 - p_1}{\rho} + g \cdot \Delta Z + \frac{u_2^2 - u_1^2}{2}$$

$$W = -827.03 + \frac{59.676 \cdot 10^5 - 0.1707 \cdot 10^5}{1000} + 9.81 \cdot 10 +$$

$$W = 5255.5158 \text{ kJ/kg}$$

4.1.5.2 The Pumps In Parallel:

- $m = m_A + m_B$
- m_a for pump 1
- m_b for pump 2

$$m = 158.8 \text{ kg/s}$$

$$(m_a) = 79.4 \text{ kg/s}, (m_b) = 79.4 \text{ kg/s}$$

$$HP_A = HP_B$$

$$H_{in} = 297.57 \text{ KJ/kg} = H_1 = H_2$$

$$T_{in} = 70^\circ \text{C}$$

For pump A:

- $H_{in}=297.57$ KJ/kg
- $\dot{M}=79.4$ kg/s
- $U_{in} = 1.5$ m /s
- $T_{in} = 70$ °C
- $P_{in} = 0.31160$ bar
- $P_{out} = 16$ bar
- $U_{out} = 1.04$ m/s
- $Z_{in} = 0$ m
- $Z_{out} = 15$ m
- $H_{out} = 858.3$ kJ/kg

$$W = \Delta H + \frac{p_2 - p_1}{\rho} + g \cdot \Delta z + \frac{u_2^2 - u_1^2}{2}$$

$$W = 858.3 - 297.57 + \frac{16 - 0.31160}{1000} + 9.81 \cdot 15 + \frac{1.04^2 - 1.5^2}{2}$$

$$W = 1166.122 \text{ kJ/kg}$$

For pump B:

- $H_{in}=297.57$ KJ/kg
- $\dot{M}=79.4$ kg/s
- $U_{in} = 1.5$ m /s
- $T_{in} = 70$ °C
- $P_{in} = 0.31160$ bar
- $P_{out} = 109.5$ bar
- $U_{out} = 1.04$ m/s
- $Z_{in} = 0$ m
- $Z_{out} = 15$ m
- $H_{out} = 1448.21$ kJ/kg

$$W = \Delta H + \frac{p_2 - p_1}{\rho} + g \cdot \Delta z + \frac{u_2^2 - u_1^2}{2}$$

$$W = 1448.21 - 297.57 + \frac{109.5 - 0.31160}{1000} + 9.81 \cdot 15 + \frac{1.04^2 - 1.5^2}{2}$$

$$W = 707.311 \text{ kJ/kg}$$

Equipment calculations

Equipment		parameter	value
Steam turbine		Work outlet	1844.57 MW
Single Pump		Specific Energy	5255.5158 kJ/kg
Pumps In Parallel	Pump A		1166.122 kJ/kg
	Pump B		707.311 kJ/kg

THE RESULTS OF THE POWER PLANT MATERIAL AND ENERGY BALANCE

Stream number	Stream description	Fluid type	Mass flowrate	P	T
			Kg/s	bara	C
1.	Exhaust gas from GT	Flue gas	539.3	1.041	519.9
2.	HP steam from HRSG#1	HP Steam	59,3	83.63	502
3.	LP steam from HRSG#1	LP Steam	20.2	7.14	220
4.	LP steam to Deaerator	LP Steam	2.8	6,55	217.6
5.	HP steam to ST	HP Steam	118,5	81	500
6.	LP steam to ST	LP Steam	37,6	6.55	217.6
7.	LLP Steam to ACC	LLP Steam	156.0	0.24	60.1
8.	Condensate from Hot Well	LP Condensate	156,0	0.57	56.7
9.	Recirculated LP Condensate	LP Condensate	156.0	4.3	56.7
10.	LP BFW to HRSGs #1&2	BFW	40.6	17.5	105.1

11.	HP BFW to HRSGs #1&2	BGW	119.1	110.5	106.5
12.	Stack Gas from HRSG #1	Flue Gas	539.3	1.01	120.6
13.	Preheated Condensate from HRSG#1	LP Condensate	78.0	1.2	94.8
14.	Preheated Condensate from HRSG #1&2 to deaerator	LP Condensate	156.0	1.2	94.8
15.	LP steam from HRSGs #1&2	LP Steam	40.4	6.55	217.6
16.	HP Blowdown from HRSG #1	Blowdown	0.3	86.6	300
17.	LP Blowdown from HRSG #1	Blowdown	0.1	7.5	168
18.	HP Blowdown from HRSG #2	Blowdown	0.3	86.6	300
19.	LP Blowdown from HRSG #2	Blowdown	0.1	7.5	168 .
20.	Exhaust gas from GT #2	Flue Gas	539.3	1.041	519.9
21.	HP steam from HRSG #2	HP Steam	59.3	83.63	502
22.	LP steam from HRSG #2	LP Steam	20.2	7.14	220
23.	Stack Gas from HRSG #2	Flue Gas	539.3	1.01	120.6
24.	Preheated Condensate from HRSG#2	LP Condensate	78.0	1.2	94.8
25.	HP BFW to HRSG #1	BFW	59.6	110.5	106.5
26.	HP BFW to HRSG #2	BFW	59.6	110.5	106.5
27.	HP BFW to Emergency use	BFW	0.8		
28.	WATER MAKE UP	DEMI WATER	20.3	3.4	15
29.	LP BFW TO HRSG #1	BFW	20.3	17.5	105.1

30.	LP BFW TO HRSG #2	BFW	0.3	17.5	105.1
31.	LIQUID DRAINAGE	WY	0.3	1	100
32.	LIQUID DRAINAGE	WY	59,3	1	100

4.2 Case 2

In the second scenario of the Combined Cycle Power Plant Material and Energy Balance, we're dealing with a change in ambient conditions. The ambient temperature is now 15°C, the pressure is 1.013 bar, and the relative humidity is 60%.

- The ambient temperature 15 C
- The ambient pressure= 1.013 bar
- The ambient relative humidity= 60%

These changes, subtle as they may seem, can have profound impacts on the performance of the power plant, particularly on the conditions of the exhaust gases entering the Heat Recovery Steam Generator (HRSG), a critical component of the power plant.

To navigate through these changes, we turn to the principles of thermodynamics and fluid dynamics, armed with equations that can help estimate the new conditions of the exhaust gases. These equations, while simplified, provide a starting point for understanding the complex interactions within the power plant.

To recalculate the conditions of the exhaust gases entering the Heat Recovery Steam Generator (HRSG), we can use the following equations:

- **Change in Temperature:** The change in exhaust gas temperature (ΔT) can be estimated based on the change in ambient temperature (ΔT_{amb}), assuming a linear relationship:

$$\Delta T = k * \Delta T_{amb}$$

where k is a constant that represents the sensitivity of the exhaust gas temperature to changes in the ambient temperature. This constant can be determined based on historical data or manufacturer's specifications.

- **Change in Pressure:** The exhaust gas pressure is typically close to the ambient pressure, so you can use the new ambient pressure as the exhaust gas pressure[12][13]

- **Change in Flow Rate:** The change in exhaust gas flow rate ($\Delta \dot{m}$) can be estimated based on the change in power output (ΔP), assuming a linear relationship:

$$\Delta \dot{m} = k' \cdot \Delta P$$

where k' is a constant that represents the sensitivity of the exhaust gas flow rate to changes in the power output.

Or we can use **the following figure** to recalculate the flowrate and temperature of the exhaust gases entering the Heat Recovery Steam Generator (HRSG): [12][14]

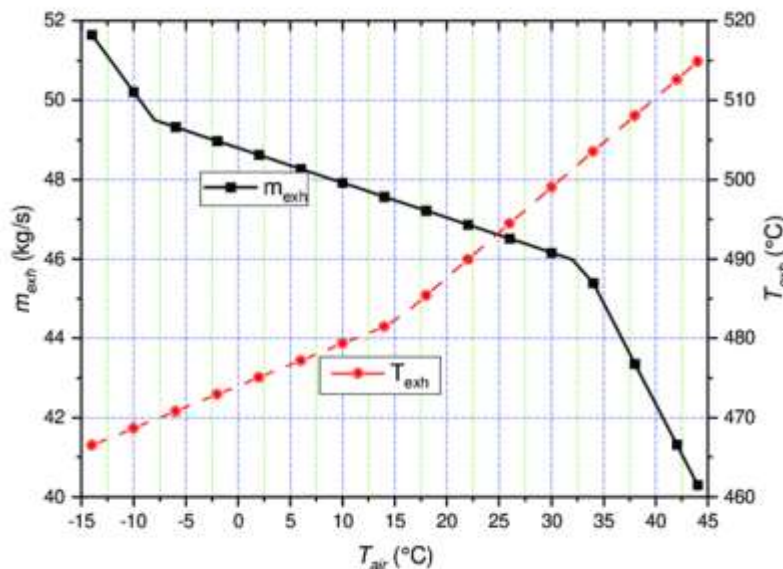


Figure 15 exhaust mass flow rate and temperature and air temperature

At $T_{air} = 15\text{ }^{\circ}\text{C}$

- $T_{exh} = 482\text{ }^{\circ}\text{C}$
- $P_{exh} = 1.013\text{ bara}$
- $m_{exh} = 47\frac{\text{kg}}{\text{s}}$

Based on these values we will recalculate:

THE RESULTS OF THE POWER PLANT MATERIAL AND ENERGY BALANCE

Stream number	Stream description	Fluid type	Mass flowrate	P	T
			Kg/s	bara	C
1.	Exhaust gas from GT	Flue gas	47	1.013	482
2.	HP steam from HRSG#1	HP Steam	59,3	106	182.04
3.	LP steam from HRSG#1	LP Steam	20.2	16	326.89
4.	LP steam to Deaerator	LP Steam	2.8	15.12	320
5.	HP steam to ST	HP Steam	118,5	101.11	182
6.	LP steam to ST	LP Steam	37,6	15.12	320
7.	LLP Steam to ACC	LLP Steam	156.0	0.4	42
8.	Condensate from Hot Well	LP Condensate	156,0	0.57	40.1
9.	Recirculated LP Condensate	LP Condensate	156.0	4.3	40.1
10.	LP BFW to HRSGs #1&2	BFW	40.6	17.5	105.1
11.	HP BFW to HRSGs #1&2	BGW	119.1	110.5	106.5
12.	Stack Gas from HRSG #1	Flue Gas	47	1.001	83

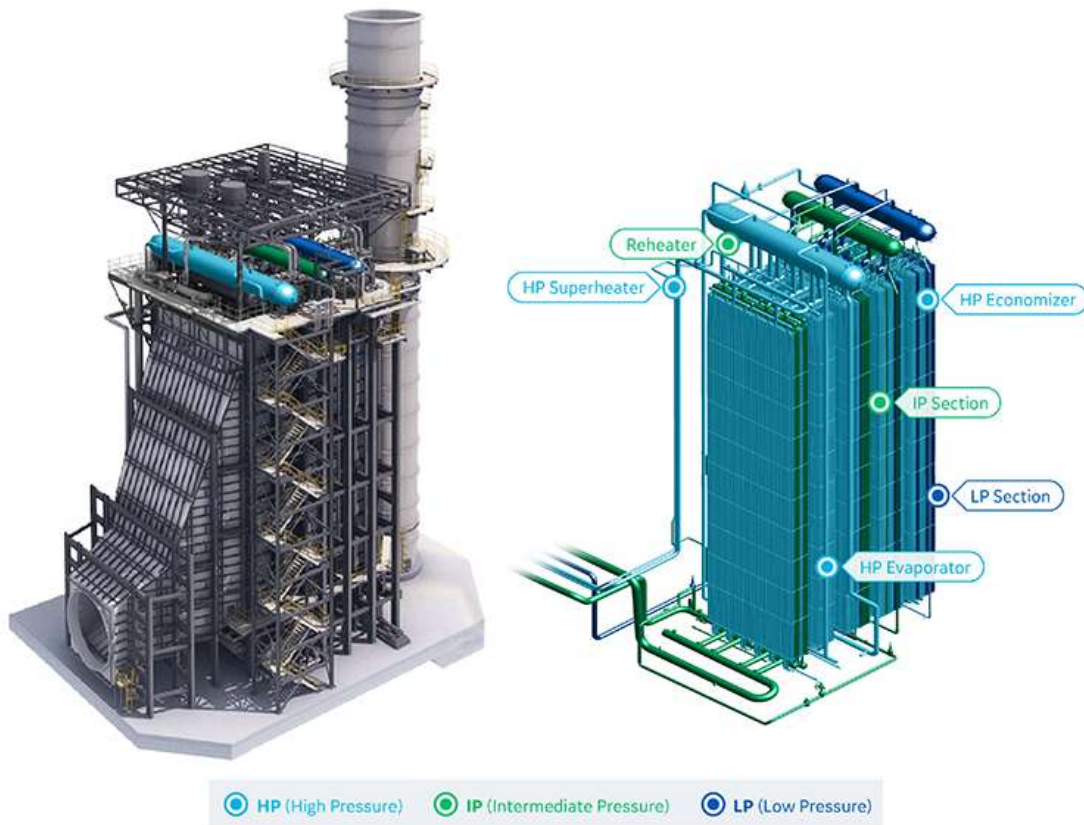
13.	Preheated Condensate from HRSG#1	LP Condensate	78.0	1.2	94.8
14.	Preheated Condensate from HRSG #1&2 to deaerator	LP Condensate	156.0	1.2	94.8
15.	LP steam from HRSGs #1&2	LP Steam	40.4	6.55	217.6
16.	HP Blowdown from HRSG #1	Blowdown	0.3	86.6	300
17.	LP Blowdown from HRSG #1	Blowdown	0.1	7.5	168
18.	HP Blowdown from HRSG #2	Blowdown	0.3	86.6	300
19.	LP Blowdown from HRSG #2	Blowdown	0.1	7.5	168 .
20.	Exhaust gas from GT #2	Flue Gas	47	1.013	482
21.	HP steam from HRSG #2	HP Steam	59.3	83.63	502
22.	LP steam from HRSG #2	LP Steam	20.2	7.14	220
23.	Stack Gas from HRSG #2	Flue Gas	47	1.001	83
24.	Preheated Condensate from HRSG#2	LP Condensate	78.0	1.2	94.8
25.	HP BFW to HRSG #1	BFW	59.6	110.5	106.5
26.	HP BFW to HRSG #2	BFW	59.6	110.5	106.5
27.	HP BFW to Emergency use	BFW	0.8		
28.	Water Make Up	DEMI WATER	20.3	3.4	15
29.	LP BFW TO HRSG #1	BFW	20.3	17.5	105.1
30.	LP BFW TO HRSG #2	BFW	0.3	17.5	105.1
31.	Liquid Drainage	WY	0.3	1	100
32.	Liquid Drainage	WY	59,3	1	100

Chapter Five

Equipment Design

EQUIPMENT DESIGN

5.1 Design of HRSG



Heat-Recover Steam Generator Component

(DATA DIMENSION SET TO START DESIGNING)

Fluids operating conditions		
Parameter	Unit	The value
$T_{gas_{in}}$	°C	519.9
$T_{gas_{out}}$	°C	120
$T_{BFW_{in}}$	°C	105

T _{BFW} _{OUT}	°C	432		
m _{gas}	kg/s	539		
m _{BFW}	kg/s	79		
HRSG Components data dimensions				
Parameter	Unit	Superheater	Evaporator	Economizer
Outer Diameter (OD)	Cm	1.905	2.54	2.54
Inner diameter (ID)	Cm	1.422	2.174	1.859
Number of fins (Nf)		4	8	8
Fin height	cm	0.635	2.007	2.007
Fin thickness	cm	0.419	0.1016	0.102
OD '(OD + 2 x height of fins)	cm	3.81	6.5532	6.5532
Thick pipe	cm	0.241	0.183	0.34
Flow area per tube	Cm	0.627	1.463	1.069
Surface per linear m (Outside)	M	0.059	0.079	0.079
Long pipe	Cm	74.98		
High Duct	M	8.99		
Duct Width	M	3.0785		
Distance between fin ends	Cm	1.27		

Design of Heat-Recovery Steam Generator Components (HRSG)

5.1.1 Calculations of Heat Transfer Area (A), Heat Duty (Q), and Heat Transfer Rate (q):

Assumed Values:

- **Specific Heat Capacities (Cp):**
 - For gas (assuming air): $Cp_{\{gas\}} = \text{approx } 1.005 \text{ kJ/kg.K}$
 - For BFW (assuming water): $Cp_{\{BFW\}} = \text{approx } 4.18 \text{ kJ/kg.K}$
- **Thermal Conductivity (k) of Tube Material:**

- For carbon steel (commonly used in HRSGs): $k_{\{tube\}} = \text{approx } 60 \text{ W/m.K}$

- **Fouling Factors:**

- Since fouling can vary widely, we'll assume typical clean conditions with
- negligible fouling resistance for this estimation.

- **Convective Heat Transfer Coefficients:**

- For gasside (assuming forced convection in a typical HRSG): $65 \text{ W/m}^2.\text{K}$
- For waterside: $6500 \text{ W/m}^2.\text{K}$

The main assumptions used to develop this mathematical model are as follows:

- Both the water and exhaust gas sides of the system are assumed to be in a steady state.
- The HRSG is unfired. There is no heat loss from the heat exchangers.
- To avoid the dew point, the stack temperature is assumed to be more than 100°C .
- The exhaust flue gases are forced to circulate using fans.
- The tubes are in a staggered arrangement with fins.
- The mass flow rate of the working fluid in the economizer, evaporator, and superheater is assumed to be equal at each pressure level. The input flue gas condition flow rate and temperature are fixed.
- Pinch and approach points have to be assumed.

Logarithmic Mean Temperature Difference (LMTD):

Calculate the temperature differences:

$$\Delta t_1 = T_{gasin} - T_{BFWout} = 519.9^\circ\text{C} - 432^\circ\text{C} = 87.9^\circ\text{C}$$

$$\Delta t_2 = T_{gasout} - T_{BFWin} = 120.6^\circ\text{C} - 105^\circ\text{C} = 15.6^\circ\text{C}$$

$$\Delta t_{LMTD} = \frac{\Delta t_2 - \Delta t_1}{\ln\left(\frac{\Delta t_2}{\Delta t_1}\right)} = \frac{15.6 - 87.9}{\ln\left(\frac{15.6}{87.9}\right)}$$

$$\Delta t_{LMTD} = -\frac{72.3}{\ln(0.177)} \approx -\frac{72.3}{-1.73} \approx 41.8^\circ C$$

Heat Duty (Q):

$$Q_{gas} = \dot{m}_{gas} \times C p_{gas} \times (T_{gas_{in}} - T_{gas_{out}}) \approx 214,775 \text{ kW}$$

Overall Heat Transfer Coefficient (U):

$$U = \left(\frac{1}{h_{gas}} + \frac{1}{h_{water}} \right)^{-1}$$

$$\approx 64.9 \text{ W/m}^2 \cdot K$$

Heat Transfer Area (A):

$$A = \frac{Q_{gas}}{U \times \Delta T_{lmLP}}$$

$$\approx 10524 \text{ m}^2$$

Heat Transfer Rate (q):

$$\begin{aligned} q &= U D \times A \times \Delta t_{LMTD} \\ &= 64.9 \text{ W/(m}^2 \cdot K) \times 10524 \text{ m}^2 \times 41.8^\circ C \\ &\approx 2.84 \times 10^7 \text{ W} \end{aligned}$$

5.1.2 The equivalent diameter for each compoenet in the HRSG.

$$A_f = \left(\frac{3.14}{4} \right) \times (OD'2 - OD2) \times 2 \times Nf \times 12$$

$$A_o = (L - Nf * y) * 3.14 * OD * 12$$

$$P = 21 * 2 * 2Nf + 2(L - Nf * Y)$$

$$de = \frac{2(Af + A_o)}{P}$$

Superheater:

Calculate A_f and A_o :

$$A_f = (3.14/4) * (3.812 - 1.92) * 2 * 4 * 12 = 34.12 \text{ cm}^2$$

$$A_o = (0.63 - 4 * 0.419) * 3.14 * 1.9 * 12 = 1.90 \text{ cm}^2$$

Calculate P :

$$P = 21 * 2 * 2 * 4 + 2 * (0.63 - 4 * 0.419) = 168.84 \text{ cm/m}$$

Calculate de :

$$de = (2 * (34.12 + 1.90)) / 168.84 = 0.43 \text{ cm}$$

Evaporator:

Calculate A_f and A_o :

$$A_f = (3.14/4) * (6.552 - 2.542) * 2 * 8 * 12 = 241.67 \text{ cm}^2$$

$$A_o = (2.007 - 8 * 0.1016) * 3.14 * 2.54 * 12 = 57.29 \text{ cm}^2$$

Calculate P :

$$P = 21 * 2 * 2 * 8 + 2 * (2.007 - 8 * 0.1016) = 336.84 \text{ cm/m}$$

Calculate de :

$$de = (2 * (241.67 + 57.29)) / 336.84 = 1.77 \text{ cm}$$

Economizer:

Calculate A_f and A_o :

$$A_f = (3.14/4) * (6.552 - 2.542) * 2 * 8 * 12 = 241.67 \text{ cm}^2$$

$$A_o = (2.007 - 8 \times 0.102) \times 3.14 \times 2.54 \times 12 = 57.15 \text{ cm}^2$$

Calculate P:

$$P = 21 \times 2 \times 2 \times 8 + 2 \times (2.007 - 8 \times 0.102) = 336.84 \text{ cm/m}$$

Calculate de:

$$d_e = (2 \times (241.67 + 57.15)) / 336.84 = 1.77 \text{ cm}$$

So, the equivalent diameters for the Superheater, Evaporator, and Economizer are 0.43 cm, 1.77 cm, and 1.77 cm respectively.

5.1.3 The Number Of Pipes Per Bundle

$$N_t = Y / ST$$

$$ST = OD + 2 \times \text{distance between the tips of the fins}$$

- N_t is the number of pipes per bundle,
- Y is the duct width, and
- ST is the vertical distance between central points of lines.
- Also, we know that:
- $ST = OD + 2 \times \text{distance between the pipes of the fins}$
- Let's calculate N_t for each component:

Superheater:

$$ST_{\text{superheater}} = 1.9 \text{ cm} + 2 + 1.27 \text{ cm} = 5.17 \text{ cm}$$

$$N_{t\text{superheater}} = 5.17 \text{ cm} / 307.85 \text{ cm} \approx 59.56$$

Since the number of pipes must be an integer, we round it down to **59 pipes**.

Evaporator and Economizer:

$$ST_{\text{evaporator/economizer}} = 2.54 \text{ cm} + 2 + 1.27 \text{ cm} = 5.81 \text{ cm}$$

$$N_{\text{evaporator/economizer}} = 5.81 \text{ cm} \times 307.85 \text{ cm} \approx 52.99$$

Since the number of pipes must be an integer, we round it down to **52 pipes**.

5.1.4 Calculation Of The Flow Area, Water Flow Speed, Mass Velocity, And Reynold Numbers In Pipes.

First, we will convert the given data to m:

- Diameters (D): 0.01422 m, 0.02174 m, 0.01859 m
- Flow areas ($\alpha't$): $6.27 \times 10^{-6} \text{ m}^2$, $1.463 \times 10^{-5} \text{ m}^2$, $1.069 \times 10^{-5} \text{ m}^2$

Superheater:

$$\alpha t = \alpha't \times Nt / 144 = 6.27 \times 10^{-6} \text{ m}^2 \times 59 / 144 = 2.59 \times 10^{-7} \text{ m}^2$$

$$Gt = W / \alpha t = 79 \text{ kg/s} / 2.59 \times 10^{-7} \text{ m}^2 = 3.05 \times 10^8 \text{ kg/(m}^2\text{.s)}$$

$$V = Gt / (3600 \times \rho_w) = 3.05 \times 10^8 \text{ kg/(m}^2\text{.s)} / (3600 \times 985 \text{ kg/m}^3) = 8.67 \text{ m/s}$$

$$Ret = D \times Gt / \mu = 0.01422 \text{ m} \times 3.05 \times 10^8 \text{ kg/(m}^2\text{.s)} / 5.1 \times 10^{-4} \text{ kg/(m.s)} = 8.53 \times 10^6$$

Evaporator:

$$\alpha t = \alpha't \times Nt / 144 = 1.463 \times 10^{-5} \text{ m}^2 \times 52 / 144 = 5.28 \times 10^{-7} \text{ m}^2$$

$$Gt = W / \alpha t = 79 \text{ kg/s} / 5.28 \times 10^{-7} \text{ m}^2 = 1.50 \times 10^8 \text{ kg/(m}^2\text{.s)}$$

$$V = Gt / (3600 \times \rho_w) = 1.50 \times 10^8 \text{ kg/(m}^2\text{.s)} / (3600 \times 985 \text{ kg/m}^3) = 4.25 \text{ m/s}$$

$$Ret = D \times Gt / \mu = 0.02174 \text{ m} \times 1.50 \times 10^8 \text{ kg/(m}^2\text{.s)} / 5.1 \times 10^{-4} \text{ kg/(m.s)} = 6.38 \times 10^6$$

Economizer:

$$\alpha t = \alpha't \times Nt / 144 = 1.069 \times 10^{-5} \text{ m}^2 \times 52 / 144 = 3.86 \times 10^{-7} \text{ m}^2$$

$$GT = W / \alpha t = 79 \text{ kg/s} / 3.86 \times 10^{-7} \text{ m}^2 = 2.05 \times 10^8 \text{ kg/(m}^2\text{.s)}$$

$$V = Gt / (3600 \times \rho_w) = 2.05 \times 10^8 \text{ kg/(m}^2\text{.s)} / (3600 \times 985 \text{ kg/m}^3) = 5.80 \text{ m/s}$$

$$Ret = D \times Gt / \mu = 0.01859 \text{ m} \times 2.05 \times 10^8 \text{ kg/(m}^2\text{.s)} / 5.1 \times 10^{-4} \text{ kg/(m.s)} = 7.47 \times 10^6$$

So, the calculated values for the superheater, evaporator, and economizer respectively are:

$$\text{Flow areas } (\alpha t): 2.59 \times 10^{-7} \text{ m}^2, 5.28 \times 10^{-7} \text{ m}^2, 3.86 \times 10^{-7} \text{ m}^2$$

$$\text{Mass velocities (Gt): } 3.05 \times 10^8 \text{ kg/(m}^2\text{.s)}, 1.50 \times 10^8 \text{ kg/(m}^2\text{.s)}, 2.05 \times 10^8 \text{ kg/(m}^2\text{.s)}$$

$$\text{Water flow speeds (V): } 8.67 \text{ m/s}, 4.25 \text{ m/s}, 5.80 \text{ m/s}$$

$$\text{Reynold numbers in pipes (Ret): } 8.53 \times 10^6, 6.38 \times 10^6, 7.47 \times 10^6$$

5.1.5 Exhaust Gas Pass Area (As)

The formula is:

$$\alpha_s = XY - (N_t \times OD \times 48) - N_t(2y \times L \times N_f \times 48)$$

For superheater:

$$\alpha_{ssh} = 0.08997 \times 0.030785 - (59 \times 0.019 / 100 \times 48) - 59 \times (2 \times 0.00419 \times 0.0063 \times 4 \times 48) = 0.00275 \text{ m}^2$$

For evaporator:

$$\alpha_{sev} = 0.08997 \times 0.030785 - (52 \times 0.0254 / 100 \times 48) - 52 \times (2 \times 0.001016 \times 0.02007 \times 8 \times 48) = 0.00267 \text{ m}^2$$

For economizer:

$$\alpha_{sec} = 0.08997 \times 0.030785 - (52 \times 0.0254 / 100 \times 48) - 52 \times (2 \times 0.00102 \times 0.02007 \times 8 \times 48) = 0.00267 \text{ m}^2$$

5.1.6 Calculating Mass Gas Flue Mass Speed (Gs)

The formula is:

$$G_s = W / \alpha_s$$

For superheater:

$$G_{ssh} = 79 / \alpha_{ssh} = 28727 \text{ kg/(m}^2 \cdot \text{s)}$$

For evaporator:

$$G_{sev} = 79 / \alpha_{sev} = 29607 \text{ kg/(m}^2 \cdot \text{s)}$$

For economizer:

$$G_{sec} = 79 / \alpha_{sec} = 29607 \text{ kg/(m}^2 \cdot \text{s)}$$

5.1.7 Determine Exhaust Gas Reynolds Numbers (Res)

The formula is:

$$Re_s = De \times G_s / \mu_s$$

For superheater:

$$Re_{ssh} = (0.019 / 100) \times G_{ssh} / (2.45 \times 10^{-5}) = 22222$$

For evaporator:

$$Re_{sev} = (0.0254/100) \times G_{sev} / (2.45 \times 10^{-5}) = 30769$$

For economizer:

$$Re_{sec} = (0.0254/100) \times G_{sec} / (2.45 \times 10^{-5}) = 30769$$

5.1.8 Fluid volumes:

Economizer:

$$V_{economizer} = N_{tubes} \times \pi \times (ID/2)^2 \times Length_{tube}$$

$$V_{economizer} = 52 \times \pi \times (1.859/2 \text{ cm})^2 \times 74.98 \text{ cm}$$

$$V_{economizer} = 10,590 \text{ cm}^3$$

Evaporator:

$$V_{evaporator} = N_{tubes} \times \pi \times (ID/2)^2 \times Length_{tube}$$

$$V_{evaporator} = 52 \times \pi \times (2.174/2 \text{ cm})^2 \times 74.98 \text{ cm}$$

$$V_{evaporator} = 14,600 \text{ cm}^3$$

Superheater:

$$V_{superheater} = N_{tubes} \times \pi \times (ID/2)^2 \times Length_{tube}$$

$$V_{superheater} = 59 \times \pi \times (1.422/2 \text{ cm})^2 \times 74.98 \text{ cm}$$

$$V_{superheater} = 59 \times \pi \times 37.86 \text{ cm}^3$$

$$V_{superheater} = 7,080 \text{ cm}^3$$

5.1.9 Total Volume of HRSG:

$$V_{HRSG} = Length_{duct} \times Width_{duct} \times Height_{duct}$$
$$20.7674 \text{ m}^3 =$$

5.1.10 HRSG Efficiency

the heat output to the gas:

$$Q_{in} = m_{gas} \times C_{p, gas} \times (T_{gas, IN} - T_{gas, OUT})$$

$$Q_{in} = 539.3 \text{ kg/s} \times 1.005 \text{ kJ/kg.K} \times ((519.9^\circ\text{C} + 273.15) - (120.6^\circ\text{C} + 273.15))$$

$$Q_{in} = 215,716.7 \text{ kJ/s}$$

the heat output to the BFW:

$$Q_{out} = m_{BFW} \times C_{p, BFW} \times (T_{BFW, OUT} - T_{BFW, IN})$$

$$Q_{out} = 79.9 \text{ kg/s} \times 4.18 \text{ kJ/kg.K} \times ((432^\circ\text{C} + 273.15) - (70^\circ\text{C} + 273.15))$$

$$Q_{out} = 120,756.5 \text{ kJ/s}$$

Calculate the efficiency of the HRSG:

$$\eta_{HRSG} = Q_{in}/Q_{out}$$

$$\eta_{HRSG} = \frac{215,716.7}{120,756.5}$$

$$\eta_{HRSG} = 0.56$$

So, the efficiency of the HRSG is approximately 0.56, or 56%. This means that 56% of the heat energy in the gas is transferred to the boiler feed water.

THE RESULTS OF THE HRSG DESIGN

Parameter	Unit	The value		
$T_{gas_{in}}$	°C	519.9		
$T_{gas_{out}}$	°C	120		
$T_{BFW_{in}}$	°C	105		
$T_{BFW_{OUT}}$	°C	432		
m_{gas}	kg/s	539		
m_{BFW}	kg/s	79		
HRSG Components design result				
Parameter	Unit	Superheater	Evaporator	Economizer
Outer Diameter (OD)	cm	1.905	2.54	2.54

Inner diameter (ID)	cm	1.422	2.174	1.859
Number of fins (Nf)		4	8	8
Fin height	cm	0.635	2.007	2.007
Fin thickness	cm	0.419	0.1016	0.102
OD '(OD + 2 x height of fins)	cm	3.81	6.5532	6.5532
Thick pipe	cm	0.241	0.183	0.34
Flow area per tube	cm	0.627	1.463	1.069
Surface per linear m (Outside)	m	0.059	0.079	0.079
Long pipe	cm	74.98		
High Duct	m	8.99		
Duct Width	m	3.0785		
Distance between fin ends	cm	1.27		
the equivalent diameters	cm	0.43	1.77	1.77
Af	cm ²	34.12	241.67	241.67
Ao	cm ²	1.90	57.29	57.17
Exhaust gas pass area (α_s)	m ²	0.00275	0.00267	0.00267
Water flow speeds (V)	m/s	8.67	4.25	5.80
Mass velocities (Gt)	kg/(m ² .s)	3.05×10^8	1.50×10^8	2.05×10^8
Reynold numbers in pipes (Ret)		8.53×10^6	6.38×10^6	7.47×10^6
The number of pipes per bundle (Nt)	pipe	59	52	52
Fluids Volume (V)	cm ³	7,080	14,600	10,590
HRSG Design result				
Total Volume (V)	m ³	20.7674		
Efficiency (η_{HRSG})	%	56		
Δt_{LMTD}	°C	41		
Heat Transfer Area (A):	m ³	10524		

5.2 Design Of Steam Turbine

Given values:

- $T_{in}=500^{\circ}C = 932^{\circ}F$
- $P_{in}=81\text{ bara} = 1,174.8 \text{ psi}$
- $h_{in} = 3489.69 \text{ kJ/kg} = 1475.79 \frac{\text{Btu}}{\text{lb}} (0.4229 \frac{\text{KJ}}{\text{KG}})$
- $\rho_{in} = 24.262 \text{ Kg/m}^3 = 1.514627 \left(\frac{\text{ft}^3}{\text{lb}} \right)$
- $h_{out} = \frac{1915 \text{ kJ}}{\text{kg}} = 809.8535 \frac{\text{Btu}}{\text{lb}}$
- $T_{out}=60.1^{\circ}C$
- $\rho_{out} = 0.161 \frac{\text{kg}}{\text{m}^3} = 0.01 \left(\frac{\text{ft}^3}{\text{lb}} \right)$
- $\dot{m}_{in} = \dot{m}_{out} = 118.5 \text{ kg/s} = 940492.777 \text{ (lb/hr)}$

5.2.1 Work:

$$\begin{aligned} W &= \dot{m}_{in} * h_{in} - \dot{m}_{out} * h_{out} \\ &= 940492.777 \text{ lb/hr} \times (1475.79 \frac{\text{Btu}}{\text{lb}} - 809.8535 \frac{\text{Btu}}{\text{lb}}) \\ W &= 6293117560 \text{ btu/hr} = 1844.57 \text{ MW} \end{aligned}$$

5.2.2 Theoretical Steam Rate

$$\begin{aligned} (\text{TRS}) &= 2545 / (h_1 - h_2), \text{ 2545 is a constant, (2545 Btu/hp-hr)} \\ \text{TRS} &= 2545 / (1436.57 - 809.8535) \\ \text{TRS} &= 4.06 \text{ (Lb/hp-hr)} \end{aligned}$$

5.2.3 Actual Steam Rate

$$(\text{ASR}) = \text{TSR} / (\text{turbine efficiency})$$

Assume, *efficiency* = 50%

$$\text{ASR} = (11.679) / 0.5 = 46.717 \text{ lb/ hp-hr, this is the actual steam rate needed}$$

5.2.4 Horsepower

$$\dot{m} = \text{ASR} \times \text{Horsepower}$$

$$940492.778 \text{ lb/hr} = 46.717 \text{ lb/ hp-hr} \times \text{Horsepower}$$

$$\text{Horsepower} = 14532.4 \text{ hp}$$

Assume, 4% Mechanical losses

5.2.5 Mechanical losses Horsepower

$$=14532.4hp \times 0.04 = 581.3hp$$

$$\text{Horsepower} = 14532.4hp + 581.3 = 15113.7hp$$

this is the power needed by the steam turbine including losses from seal and bearings.

$$\begin{aligned} \text{Steam power} &= (h_1 - h_2) \times \text{Steam Flow Rate}/w \\ &= (1475.79 \frac{Btu}{lb} - 809.8535 \frac{Btu}{lb}) \times 940492.778 \text{ lb}/ \\ &\text{hr} \div 96563608218 \text{ Btu/hr} \\ &= 6.48597 \times 10^3 \end{aligned}$$

5.2.6 Shaft power

$$\begin{aligned} &= \text{steam power} - \text{mechanical losses} \\ &= 6.48597 \times 10^3 - 581.3hp \\ &= 5904.67hp = 4403kw \end{aligned}$$

Q volumetric flow rate

$$\begin{aligned} \frac{m}{\rho} &= (118.5 \text{ kg/s}) / 24.262 \text{ Kg/m}^3 = 4.884 \frac{m^3}{s} \\ &= 620916.9 \text{ ft}^3/\text{hr} \end{aligned}$$

5.2.7 Pipe Size for the Inlet and Exhaust

Diameter = [(Steam weight flow (lb/hr) × specific volume (ft³ /lb) × 0.051) / steam velocity (ft /sec.)]^{1/2}

$$\begin{aligned} \text{specific volume} \left(\frac{ft^3}{lb} \right) &= \frac{1}{\text{density}} \\ v_{in} &= 1 / 1.514627 \left(\frac{ft^3}{lb} \right) = 0.66 \frac{ft^3}{lb} \\ v_{out} &= \frac{1}{0.01} = 100 \frac{ft^3}{lb} \end{aligned}$$

Assume

1. Maximum recommended Inlet steam velocity = 175 feet/second
2. Maximum recommended Exhaust steam velocity = 250 feet/second

$$D_{in} = [(940492.778 \text{ lb/hr} \times 0.66 \text{ ft}^3 / \text{lb} \times 0.051) / 175]^{1/2} = 13.5 \text{ in}$$

Now calculate the size of the exhaust pipe.
inside diameter of the pipe needed.

$$D_{ex} = \left[\frac{940492.778 \left(\text{lb} \frac{\text{lb}}{\text{hr}} \right) \times 100 \left(\frac{\text{ft}^3}{\text{lb}} \right) \times 0.051}{250 \left(\frac{\text{ft}}{\text{sec}} \right)} \right]^{1/2} = 138.5 \text{ in}$$

5.2.8 Shaft Size

$$d = \left[\frac{321,000 \times \text{Power}}{\text{Shear stress} \times \text{shaft speed}} \right]^{1/3} \text{ or } d = \left[\frac{321,000 \times P}{S_s \times N} \right]^{1/3}$$

$$D = 4.7 \text{ inch}$$

d = shaft diameter (inches)

P = shaft horsepower (hp) = 5904.67 hp

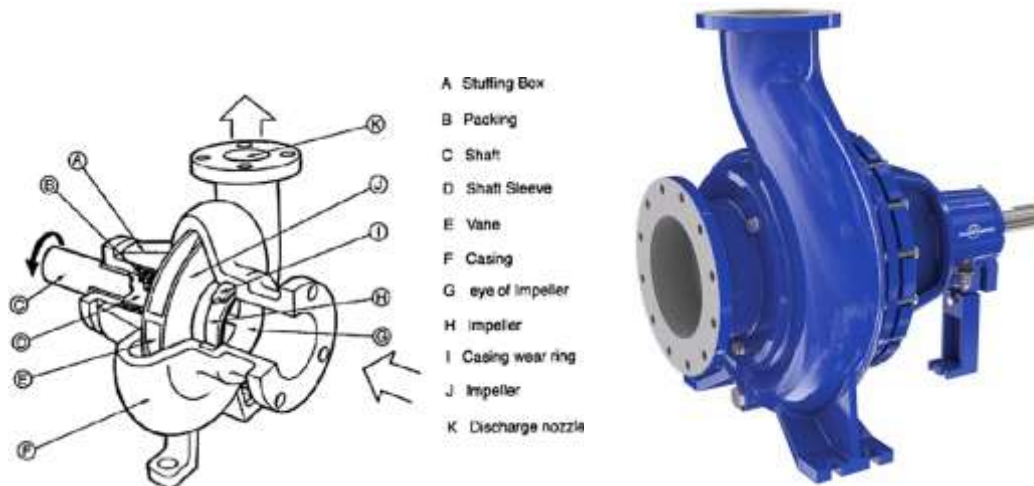
S_s = Shear Stress allowed in shaft material (psi) assume, 5000 psi

N = shaft speed in rpm assume, 3600 rpm

$$D = 4.7 \text{ inch}$$

Designation	Material	Maximum shear stress Ss (psi)
AISI -1040	Medium Carbon Steel	5000
AISI 4140	Chrome Moly Alloy	11500
AISI 4340	Nickel Chrome Moly Alloy	12500

5.3 Design Of Single Section Centrifugal Pump



Single Section Centrifugal Pump

Operation condition

- $T = 55$
- $P = 1.007 \text{ bar}$
- At this condition the water properties are
- $\rho = 985.65 \text{ kg/m}^3$
- $\mu = 0.5030 \times 10^{-3} \text{ Pa}\cdot\text{s}$

For the section

- $D_i=12$
- $U=1.5$ m/s
- $=4.572 \cdot 10^{-5}$ m for steel or iron pipe

$$\text{Re.no} = \frac{\rho \cdot u \cdot D}{\mu}$$

$$\text{Re} = (985.65 \cdot 1.5 \cdot 12) / (0.5030 / 1000)$$

$$\text{Re} = 35271769.3 > 4000$$

The flow is turbulent

if the Reynolds number is below 2000 than the flow is said to be in a laminar regime. If the Reynolds number is above 4000 the regime is turbulent. The velocity is usually high enough in industrial processes and homeowner applications to make the flow regime turbulent.

$$f = \frac{0.25}{\left(\log_{10} \left(\frac{\epsilon}{3.7D} + \frac{5.71}{\text{Re}^{0.9}} \right) \right)^2}$$

$$f = \frac{0.25}{\left(\log_{10} \left(\frac{4.572 \cdot 10^{-5}}{3.7 \cdot 12} + \frac{5.71}{35271769.3^{0.9}} \right) \right)^2}$$

$$= 7.083 \cdot 10^{-3}$$

from darcy equation

assum that the pipe length equal to 20 m

$$hf = \frac{f \cdot l \cdot u^2}{2 \cdot D \cdot g}$$

$$hf_s = \frac{7.083 \cdot 10^{-3} \cdot 20 \cdot 1.5^2}{2 \cdot 12 \cdot 9.81}$$

$$hfs = 1.3537 \cdot 10^{-3}$$

For Discharge

- $D_{in} = 10$ m
- $u = 2.16$ m/s

$$\text{Re.no} = \frac{\rho \cdot u \cdot D}{\mu}$$

$$\text{Re} = (985.65 \cdot 2.61 \cdot 10) / (0.5030 \cdot 10^{-3})$$

$$\text{Re} = 51144065$$

With the same equations

- $f = 7.6342 \times 10^{-3}$
- $h_{fd} = 3.630 \times 10^{-3} \text{ m}$
- $p_s = 0.1707 \times 10^5 \text{ pa}$
- $p_d = 59.676 \times 10^5 \text{ pa}$
-

$$= \Delta H = \left(\frac{P_d - P_s}{\rho g} \right) + z_d - z_s + (h_{fd} - h_{fs})$$

$$= \Delta H = \left(\frac{59.676 \times 10^5 - 0.1707 \times 10^5}{985.65 \times 9.81} \right) + 15 - 0 + (3.630 \times 10^{-3} - 0)$$

$$\Delta H = 630 \text{ m}$$

$$\text{NPSH} = \frac{P_s - p^\circ}{\rho g} + z_s - h_{fs}$$

$$P^\circ \text{ of water at } 56^\circ\text{C} = 15732 \text{ pa}$$

$$\text{NPSH} = \frac{0.1707 \times 10^5 - 15732}{985.65 \times 9.81} + 0 - 1.3537 \times 10^{-3}$$

$$\text{NPSH at } z_s = 0 \text{ equal to } 0.137$$

Assumed that the

Depending upon specific speed (N_s) or total head (H_{mano}) the inlet diameter D_1 is kept in range;

$D_1 = 1/3 D_2$ to $2/3 D_2$. An average value of $D_1 = 0.5 D_2$, is usually taken.

$$D_1 = 0.5 D_2$$

$$D_2 = \frac{84.6 K_u \sqrt{H_{mono}}}{N}$$

K_u , varies from 0.95 to 1.25

K_f , varies from 0.1 to 0.25

$$H_{monometric} = H_{static} - \text{losses in pipe} + \frac{u_d^2}{2g}$$

$$(h_s - h_d) + (h_{fs} + h_{fd}) + \frac{u_d^2}{2g}$$

5.3.1 Minimum Speed For Starting A Centrifugal Pump

Assuming $D_1=0.5D_2 \therefore u_1 = 0.5 u_2$

$$Ku = \frac{u_2}{\sqrt{2gH_{mono}}}$$

rang Ku= 0.95-1.25

take 1.1

$$1.1 = \frac{u_2}{\sqrt{2 \cdot 9.81 \cdot 619.8}}$$

$$U_2 = 121.2 \text{ m/s}$$

$$U_1 = 0.5 u_2 = 60.6$$

$$\frac{u_2 - u_1}{2g} = H_{mono}$$

$$\frac{u_2 - 0.5 u_2}{2g} = H_{mono}$$

$$0.5 u_2 = 2g H_{mono}$$

$$U_2 = \frac{\pi D_2 N}{60}$$

$$\frac{\pi D_2 N}{60} = 4g H_{mono}$$

Assum that N equail to 1200 r.p.m

$$\frac{\pi D_2 1200}{60} = 4g H_{mono}$$

$$D_2 = 386.4 \text{ m}$$

$$D_1 = 386.4 \cdot 0.5 = 193.2 \text{ m}$$

Since the intery fluid is radial

$$V_{wl} = 0$$

Assume that the angels

$$\beta_1 = 0, \beta_2 = 35^\circ$$

$$V_f = K_f * \sqrt{2 g H_{mono}}$$

Kf rang 0.1-0.25

Take 0.11

$$V_f = 0.11 * \sqrt{2 * 9.81 * 618.86}$$

$$V_{f2} = 1114.2 \text{ m}$$

$$\tan \beta = \frac{V_{f2}}{V_{w2}}$$

$$\tan 35 = \frac{1114.2 \text{ m}}{V_{w2}}$$

$$V_{w2} = 1591.2 \text{ m}$$

$$\tan \phi = \frac{V_{f2}}{u_2 - V_{w2}}$$

$$\tan \phi = \frac{1114.2}{121.2 - 1591.2}$$

$$H_{monometric} = H_{static} - \text{loss in pipe} + \frac{u_d^2}{2g}$$

$$(h_s - h_d) + (h_{fs} + h_{fd}) + \frac{u_d^2}{2g}$$

$$H_{mono} = 618.86 \text{ m}$$

$$Q = U * A$$

$$Q_{in} = 2.16 * 113.09$$

$$Q_{in} = 169.64 \text{ m}^3/\text{s} = Q_{out}$$

$$Q_s = Q/60 = 2.827 \text{ M}^3/\text{S}$$

$$N_s = \frac{1N * \sqrt{1Q}}{H^{\frac{3}{4}}}$$

$$N_s = \frac{1200 * \sqrt{169.64}}{618^{\frac{3}{4}}}$$

$$N_s = 125.96 \text{ r. p. m}$$

Input power of centrifugal pump can be determined by following equation.

$$L = \frac{\rho Q_s g H}{\eta}$$

$$L = \frac{985.65 * 2.827 * 9.81 * 618.86}{1}$$

16916478.27 kw

For charge condition of the pump work, maximum shaft power or rated output of an electric motor L_r (kW) is decided by using Equation (4).

$$L_r = \frac{(1+Fa)*L}{\eta_{tr} * 1000}$$

Where, Fa is the allowance factor, and 0.1~ 0.4 for an electric motor and larger than 0.2 for engines And then, η_{tr} is the transmission efficiency, and 1.0 for direct coupling and 0.9 ~ 0.95 for belt drive. The shaft diameter at hub section of impeller is

$$L_r = \frac{(1+Fa)*L}{\eta_{tr} * 1000}$$

$$L_r = \frac{(1+0.3)*16916478.27}{0.9*1000}$$

$$L_r = 24434.91 \text{ kw}$$

$$T = 60 L_r / 2\pi N$$

$$60 * 24434.91 / 2\pi * 1200$$

$$T = 194.44$$

$$K_{mo} = (0.07 \sim 0.11) +$$

$$K_{mo} = (0.08 + (0.00023 * 125.96))$$

$$K_{mo} = 0.0921$$

The volute is designed by determining the angle Φ° measured from and assumed radial line by tabular integration of E

$$\Phi^\circ = \frac{132 \log_{10} R_t / R_2}{\tan \alpha}$$

$$\Phi^\circ = 30^\circ$$

Pump Data Sheet			
N0	description	Symbols	Result
1	Input power	l	16916478.27 kw
2	section nozzle diameter	D _{in}	12 m
3	Discharge nozzle diameter	D _{out}	10 m
4	Impuler inlet diameter	D ₁	193.2 m
5	Impuler outlet diameter	D ₂	386.4 m
6	Inlet angle of impeller blade	β_1	0°
7	outlet angle of impeller blade	β_2	35°
8	Tangential velocity of imp inlet	U ₁	60.6 m/s
9	Tangential velocity of imp outlet	U ₂	121.2 m/s
10	Volute tongue angle	Φ°	= 30°

5.4 ACC Design

Assumed data:

- Number of rows of the exchanger (Nr): 5
- Fins density: 394 fins/m
- Tube spacing: 0.0635 m
- Face velocity (FV): 3.18 m/s

Data required for the design

- Type of heat exchanger: air cooled
- Tube material: stainless steel
- Air ambient temperature (T₁): 35 c = 308 k
- Hot fluid to cool: steam

- Hot fluid flow rate (ms): 156 kg/s
- Hot fluid specific heat (Cp): 1.005 kJ/kg.k
- Hot fluid inlet temperature (t1): 60.1 c = 333.1 k
- Hot fluid outlet temperature (t2): 56.7 c = 329.7 k
- Tube inside diameter (Di): 0.0211 m
- Tube outside diameter (Do): 0.0254 m

The Air-Side Heat Transfer Coefficient

depending on the density of fins on the tubes (394 fins/m):

$$h_a = 8 \times (FV)^{1/2}$$

$$h_a = 8 \times (3.18)^{1/2} = 14.3 \text{ w/m}^2.\text{k}$$

heat transfer coefficient through the tube

*Tube thermal conductivity of the tube material ($\lambda_w = k$): pproax 43 w/m.k

$$h_w = 2 \times \lambda_w / (D_o - D_i)$$

$$h_w = 2 \times 43 / (0.0254 - 0.0211) = 20000 \text{ w/m}^2.\text{k}$$

5.4.1 The Overall Heat Transfer Coefficient (U)

$$\frac{1}{U} = \frac{1}{h_a} + \frac{1}{h_i \left(\frac{D_i}{D_o} \right)} + \frac{1}{h_w} + \frac{1}{h_s}$$

*we'll assume typical clean conditions with negligible fouling resistance for this estimation.

$$Re_i = \frac{D_i \rho_i v_i}{\mu_i}$$

Where, (Di) inside diameter, (ρ_i) is fluid density, (v_i) velocity, (μ_i) viscosity,

$$Re_i = \frac{0.0211 \times 1.0586 \times 3.18}{1.9977 \times 10^{-5}} = 3555$$

For $Re_i > 2,100$

From the table prandtl number (Pri) = 0.701

*the length of the heat exchanger (L) has a typical maximum of (14m) 48ft.

$$Nu_i = 0.116 (Re_i^{2/3} - 125) Pr_i^{1/3} [1 + (D_i/L)^{2/3}]$$

Where, (Nui) is nusselt number

$$Nui = 0.116(3555^{2/3} - 125) \times 0.701^{1/3} [1 + (0.0211/12)^{2/3}]$$

$$Nui = 11.28$$

$$hi = (\lambda m/Di) \times Nui = (43/0.0211) \times 11.28 = 22987 \text{ w/m}^2\text{k}$$

$$\frac{1}{U} = \frac{1}{14.3} + \frac{1}{22987 \left(\frac{0.0211}{0.0254} \right)} + \frac{1}{20000} = 14.24 \text{ w/m}^2\text{k}$$

From the table: the air outlet temperature (T2) = 83c = 356k

Size of the air cooled heat exchanger

$$Q = ms \times cp \times (t2 - t1) = 156 \times 1.005 \times (329.7 - 333.1) = 46391 \text{ kj/s}$$

$$FA = Q/[FV \times (T2 - T1) \times 1.95]$$

(FA) is the face area of heat exchanger

$$FA = 46391/[3.18 \times (356 - 308) \times 1.95] = 155 \text{ m}^2$$

The heat exchanger width can be calculated from the following formula:

$$Y = \text{width} = FA/L = 155/12 = 12.9\text{m} \cong 13\text{m}$$

the mean temperature difference (ΔT_m)

$$\Delta T_m = \frac{[(T2 - T1) - (t1 - t2)]}{\ln \left[\frac{(T2 - T1)}{(t1 - t2)} \right]} = \frac{[(356 - 308) - (333.1 - 329.7)]}{\ln \left[\frac{(356 - 308)}{(333.1 - 329.7)} \right]} = 16.85 \text{ k}$$

the area required for reaching the exchange duty under this mean temperature difference (Acalculated)

$$A_{\text{calculated}} = Q/(U \times \Delta T_m) = 46391/(14.24 \times 16.85) = 193\text{m}^2$$

the available area (Aavailable)

$$A_{\text{available}} = N_r \times N_t \times \pi \times Do \times L$$

Where (Nt) is number of tubes per row which can be calculated with the following equation:

$$N_t = Y/\text{pitch} = 13/0.0635 = 204.7$$

$$A_{\text{available}} = 5 \times 204.7 \times 3.14 \times 0.0254 \times 12 = 979.6 \text{ m}^2$$

*If the area available is > area calculated (required) then the design is acceptable.

5.4.2 The Air Side Pressure Drop And Fan Power requirement

$$\Delta P_a = 0.0037 \times N_r \times (FV/100)^{1.8}$$

$$\Delta P_a = 0.0037 \times 5 \times (3.18/100)^{1.8} = 3.73 \times 10^{-5} \text{ pa}$$

$$b_{hp} = FV \times F_A \times (T_2 + 273.15) \times (\Delta P_a + 0.1) / (1.15 \times 10^6)$$

$$b_{hb} = 3.18 \times 155 \times (356 + 273.15) \times (3.73 \times 10^{-5} + 0.1) / (1.15 \times 10^6) = 1983.56 \text{ w}$$

Table 1A. Properties of saturated steam (SI units)

Absolute pressure (kN/m ²)	Temperature		Enthalpy per unit mass (H_f)			Entropy per unit mass (S_f)			Specific volume (v)	
	(°C)	(K)	(kJ/kg)			(kJ/kg K)			(m ³ /kg)	
	t_s	T_s	water	latent	steam	water	latent	steam	water	steam
<i>Datum:</i> Triple point of water										
0.611	0.01	273.16	0.0	2501.6	2501.6	0	9.1575	9.1575	0.0010002	206.16
1.0	6.98	280.13	29.3	2485.0	2514.4	0.1060	8.8706	8.9767	0.001000	129.21
2.0	17.51	290.66	73.5	2460.2	2533.6	0.2606	8.4640	8.7246	0.001001	67.01
3.0	24.10	297.25	101.0	2444.6	2545.6	0.3543	8.2242	8.5785	0.001003	45.67
4.0	28.98	302.13	121.4	2433.1	2554.5	0.4225	8.0530	8.4755	0.001004	34.80
5.0	32.90	306.05	137.8	2423.8	2561.6	0.4763	7.9197	8.3960	0.001005	28.19
6.0	36.18	309.33	151.5	2416.0	2567.5	0.5209	7.8103	8.3312	0.001006	23.74
7.0	39.03	312.18	163.4	2409.2	2572.6	0.5591	7.7176	8.2767	0.001007	20.53
8.0	41.54	314.69	173.9	2403.2	2577.1	0.5926	7.6370	8.2295	0.001008	18.10
9.0	43.79	316.94	183.3	2397.9	2581.1	0.6224	7.5657	8.1881	0.001009	16.20
10.0	45.83	318.98	191.8	2392.9	2584.8	0.6493	7.5018	8.1511	0.001010	14.67
12.0	49.45	322.60	206.9	2384.2	2591.2	0.6964	7.3908	8.0872	0.001012	12.36
14.0	52.58	325.73	220.0	2376.7	2596.7	0.7367	7.2966	8.0333	0.001013	10.69
16.0	55.34	328.49	231.6	2370.0	2601.6	0.7721	7.2148	7.9868	0.001015	9.43
18.0	57.83	330.98	242.0	2363.9	2605.9	0.8036	7.1423	7.9459	0.001016	8.45
20.0	60.09	333.24	251.5	2358.4	2609.9	0.8321	7.0773	7.9094	0.001017	7.65
25.0	64.99	338.14	272.0	2346.4	2618.3	0.8933	6.9390	7.8323	0.001020	6.20
30.0	69.13	342.28	289.3	2336.1	2625.4	0.9441	6.8254	7.7695	0.001022	5.23
35.0	72.71	345.86	304.3	2327.2	2631.5	0.9878	6.7288	7.7166	0.001025	4.53
40.0	75.89	349.04	317.7	2319.2	2636.9	1.0261	6.6448	7.6709	0.001027	3.99
45.0	78.74	351.89	329.6	2312.0	2641.7	1.0603	6.5703	7.6306	0.001028	3.58
50.0	81.35	354.50	340.6	2305.4	2646.0	1.0912	6.5035	7.5947	0.001030	3.24
60.0	85.95	359.10	359.9	2293.6	2653.6	1.1455	6.3872	7.5327	0.001033	2.73
70.0	89.96	363.11	376.8	2283.3	2660.1	1.1921	6.2883	7.4804	0.001036	2.37
80.0	93.51	366.66	391.7	2274.0	2665.8	1.2330	6.2022	7.4352	0.001039	2.09
90.0	96.71	369.86	405.2	2265.6	2670.9	1.2696	6.1258	7.3954	0.001041	1.87
100.0	99.63	372.78	417.5	2257.9	2675.4	1.3027	6.0571	7.3598	0.001043	1.69

The following table can be used to estimate the air outlet temperature :

Process inlet temperature, c	Outlet air temperature, c		
	U=50 (BTU/h.ft ² .F)	U=100 (BTU/h.ft ² .F)	U=150 (BTU/h.ft ² .F)
175	90	95	100
150	75	80	85
125	70	75	80
100	60	65	70
90	55	60	65
80	50	55	60
70	48	50	55
60	45	48	50
50	40	41	42

(This table is based on an ambient temperature in between 32 and 37c)

Chapter Six

Cost Estimation

COMBINED CYCLE POWER PLANT COST ESTIMATION

6.1 Introduction:

In the context of constructing a **combined cycle power plant**, the significance of cost estimation cannot be overstated. Accurate cost estimation serves as the backbone of the project's financial structure, playing a pivotal role in budgeting and planning. It provides a comprehensive financial blueprint that guides the allocation of resources, ensuring that the project adheres to its financial constraints. By predicting the expenses associated with each phase of the project, from procurement of materials to labor costs, cost estimation aids in avoiding financial overruns, thereby ensuring the project's economic feasibility and success. Thus, cost estimation is not merely a step in the project lifecycle, but a critical tool for effective project management.[15]

6.2 Factors Affecting Cost:

The cost of constructing a **combined cycle power plant** is influenced by a multitude of factors, each contributing to the overall financial blueprint of the project.

1. **Equipment Costs:** This includes the cost of procuring and installing all necessary machinery and technology. The type and scale of the equipment required can significantly impact the total cost.

2. **Labor Costs:** The expenses associated with hiring skilled labor for the construction and operation of the plant. This can vary based on the location, duration of the project, and the expertise required.
3. **Land Acquisition Costs:** The cost of acquiring suitable land for the power plant. This can fluctuate greatly depending on the location and size of the land.
4. **Regulatory Requirements:** Compliance with local and international regulations can add to the cost. This includes safety standards, environmental regulations, and any permits or licenses required.
5. **Additional Expenses:** These can encompass a wide range of costs such as environmental mitigation measures, infrastructure development, contingency costs, and any unforeseen expenses that may arise during the project.

Each of these factors must be meticulously estimated and accounted for in the project's budget to ensure its financial viability and [success](#).[\[15\]](#) [\[17\]](#)

6.3 The Cost Estimation Method

For the cost estimation of the combined cycle power plant, we will employ a **bottom-up approach**. This method involves breaking down the project into smaller components, estimating the cost of each component, and then summing these costs to arrive at the total cost.

The total cost (T) can then be calculated by summing up all these costs:

$$T = E + L + La + R + A$$

Where:

- **Equipment Costs (E):** We will list all the necessary equipment and machinery required for the power plant. For each item, we will estimate the cost, including procurement and installation. The total equipment cost will be the sum of the costs of all individual items.
- **Labor Costs (L):** We will estimate the number of workers required, their wages, and the duration of their work. The labor cost will be the product of the number of workers, their wages, and the duration of work.
- **Land Acquisition Costs (La):** We will estimate the cost of the land based on its location and size.
- **Regulatory Costs ®:** We will identify all the necessary permits and licenses and estimate their costs.

- **Additional Expenses (A):** We will identify any additional costs such as environmental mitigation measures, infrastructure development, and contingency costs.

6.4 The estimating process:

1. Equipment Costs:

Total equipment cost = \$50M +(2 * \$40M) + \$25M + \$15M = **\$170** million
[17]

EQUIPEMNT	Configuration	Approximate Price
Steam Turbine (ST)	Multi-pressure condensing type.	\$30 million - \$60 million
Heat Recovery Steam Generator (HRSG)	Dual-pressure boiler	\$25 million - \$50 million
Deaerator	Typically straightforward design.	\$25 milion
Air Cooler Condenser (ACC)	Air-cooled condenser	\$10 million - \$20 million

2. Labor Costs:

By assuming we need 100 workers for 2 years, each earning \$50 per hour on average.

Total labor cost = 100 workers * 40 hours/week * 52 weeks/year * 2 years * \$50/hour = \$20,800,000

Number of workers	years	the average hourly wage	Total labor cost
100	2 hours /week	50 \$/hour	\$20,800,000

3. Land Acquisition Costs:

- Assuming that the land purchase price is \$5 million, Include additional expenses such as land surveys (\$50,000) and legal fees (\$100,000).

Total land acquisition cost = \$5,000,000 + \$50,000 + \$100,000 = \$5,150,000

4. Regulatory Requirements:

- Estimating the costs associated with obtaining permits and approvals, Let's say permit fees and other regulatory expenses amount to \$1 million.

5. Additional Expenses:

- By allocating \$2 million for infrastructure development and \$1 million for contingency.

By suming up all the costs:

Total Cost = Equipment Costs + Labor Costs + Land Acquisition Costs + Regulatory Requirements + Additional Expenses

Total Cost = \$170,000,000 (Equipment) + \$20,800,000 (Labor) + \$5,150,000 (Land) + \$1,000,000 (Regulatory) + \$2,000,000 (Additional) + \$1,000,000 (Contingency)

Equip ment	Labor	Land	Regul atory	Addit ional	Contin gency	Total Cost
\$170,00 0,000	\$20,80 0,000	\$5,15 0,000	\$1,000 ,000	\$2,00 0,000	\$1,000, 000	\$199,95 0,000

So, the estimated total cost of the combined cycle power plant is **approximately \$199.95 million**, and with **engineering cost 10%** of total cost, **total cost= \$19,995,000**

Chapter Seven

Conclusion & Recommendations

CONCLUSION & RECOMMENDATIONS

7.1 Conclusion:

The study has demonstrated that the combined cycle is applicable to the selected gas power station, with an energy increase of no less than 50%. Furthermore, the closed cycle has proven to reduce pollution and thermal emissions, converting them into electrical energy. The inlet temperature of exhaust gases coming from gas turbines has decreased to 120°C from 520°C (ambient temperature of 55°C).

7.2 Recommendations:

1. We recommend conducting a study to explore the feasibility of using the closed cycle for power generation and desalination of seawater.
2. Investigate the possibility of recovering carbon dioxide emissions from the chimney to the closed cycle and producing ethanol fuel for use as a secondary fuel instead of diesel, alongside gas, for power generation in dual-phase turbines.
3. Study the utilization of water vapor and recovered carbon dioxide from chimneys in injection processes to enhance reservoir pressure underground.
4. Consider the utilization of solar energy for heat generation and water heating to produce electrical energy in the steam turbine.

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